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IRON AND STEEL ENGINEER

NOVEMBER • 1940

VOL. XVII NO. XI

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Published Monthly by

ASSOCIATION OF IRON AND STEEL ENGINEERS

Entered as second-class matter January 25th, 1924, at Pittsburgh, Pa., under the Act of March 3rd, 1879

Statements and opinions given in articles and papers appearing in "Iron and Steel Engineer" are the expressions of contributors, for which the Association of Iron and Steel Engineers assumes no responsibility.

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Single Copy Foreign Prepaid $2.00 Single Copy $1.50

Subscription price in U. S. & Canada ..... $ 7.50 per year

Subscription price in foreign countries ..... $10.00 per year

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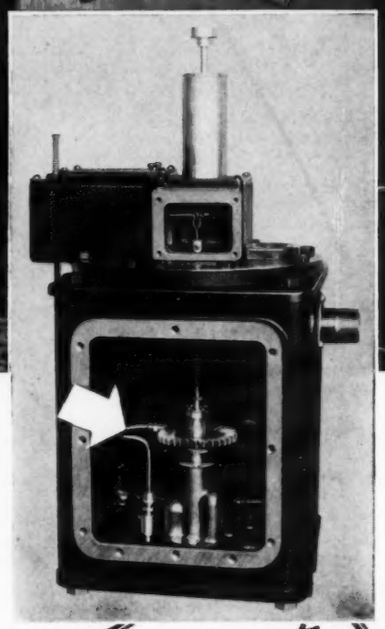

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Trends in Steel Mill

POWER GENERATION

By C. C. FRANCK

Turbine Engineering Department

WESTINGHOUSE ELECTRIC & MANUFACTURING CO.

PHILADELPHIA, PENNA.

Presented before A. I. S. E.

Birmingham Section, November 27, 1939

▲ IN considering this subject, one might be inclined to think solely in terms of charts and diagrams, but with the wealth of subject matter created by the common problems of the steel mills and a manufacturer of steam turbines, the preparation presented a problem of how to keep the length within the bounds of a single paper. We, as manufacturers of steam turbines are indebted to the steel manufacturers for improvements in materials which make it possible for us to establish "Trends in Power Generation."

Trends in power generation can best be studied by referring to Figure 1, which covers the subject from 1900 to the present day. The trend in the type and size of units is particularly interesting, starting with 1800 rpm. units at ratings of 1000 kw. we note that sizes increased to about 10,000 kw. in 1908. Temperatures and pressures were low and the machines were for the greater part constructed of cast iron material. The widespread use of cast and forged steel parts made possible an increase in temperature and pressure, and greater allowable stresses provided for increase in physical size of the turbines. Consequently, in 1920 to 1922 the size of units had increased to 60,000 kw. and pressures of 250 lb. per sq. in. and 550 degrees F. total temperature were in common use. At about this period, the widespread demand for industrial turbines of from 1000 to 7500 kw. firmly established the 3600 rpm. type unit, the value of which in decreasing the physical size was unquestioned. For the greater part, units of 10,000 kw. and above were confined to the 1800 rpm. class and by various arrangements of turbines, ratings in 1928 were increased to 165,000 kw. Steam conditions on the average were in the vicinity of 400 lb. per sq. in. and 700 degrees F.

Thermal studies indicated that the use of increased pressure and temperature would result in tremendous operating economies, but since materials limited the upper temperatures, it was only possible to utilize the increase in pressure. Steam conditions of 1250 lb. per sq. in., 750 degrees F. total temperature would naturally

FIGURE 1—Chart showing the progress in certain items of turbine design since 1900.

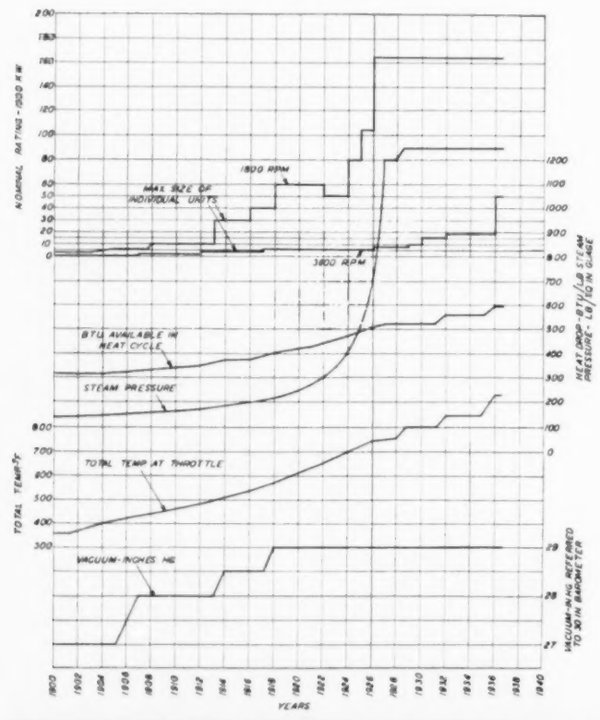
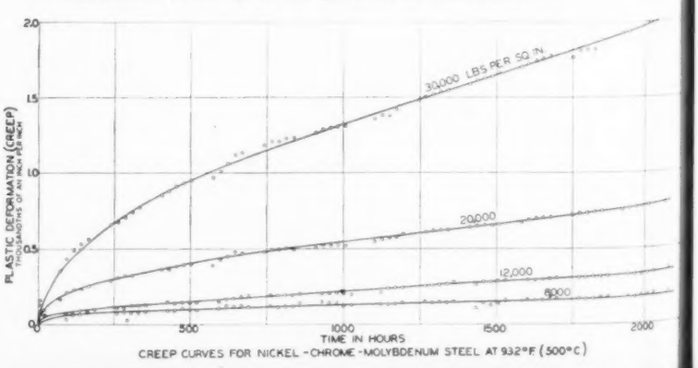


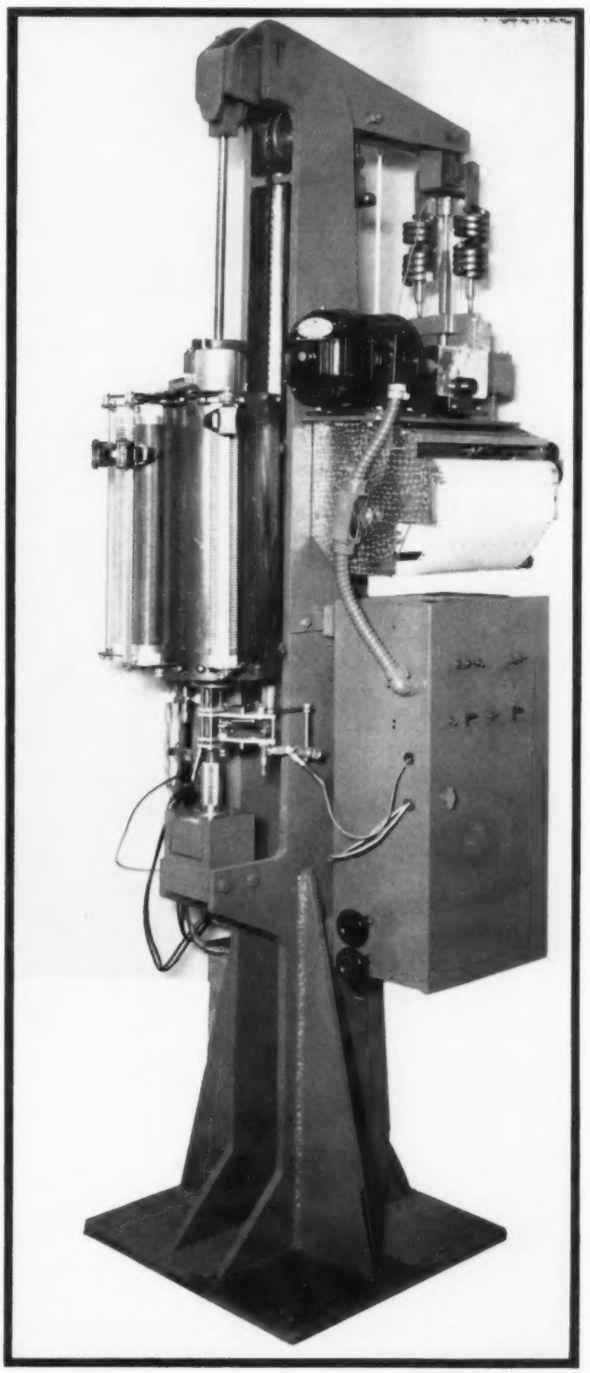
FIGURE 2—Results of "creep" tests show the great influence of increased stress upon the permanent deformation.



CREEP CURVES FOR NICKEL-CHROME-MOLYBDENUM STEEL AT 932°F (500°C)

be accompanied by excessive moisture and erosion problems in the exhaust blading of a straight through expansion turbine. To combat this it was necessary to resort to interstage reheating, and boiler complications as well as plant layout problems presented a handicap to this part of the development. The stations using this principle were highly successful and it is to be noted that the fore-runner of the present "topping" turbines developed at this time. These units were of the 3600 rpm. class, about 10,000 to 12,500 kw. in capacity, and operating with steam conditions of 1200 lb. per sq. in., 700 to 775 degrees F., and exhausting into the reheater at 100 to 150 lb. per sq. in.

FIGURE 3—This machine records the "relaxation," or variation, with time and temperature, of stress established in the test piece.



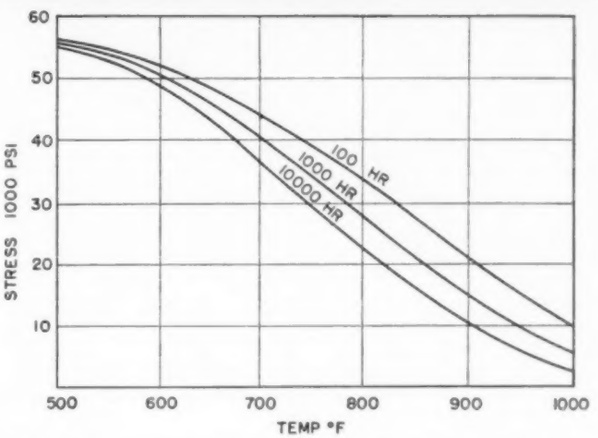
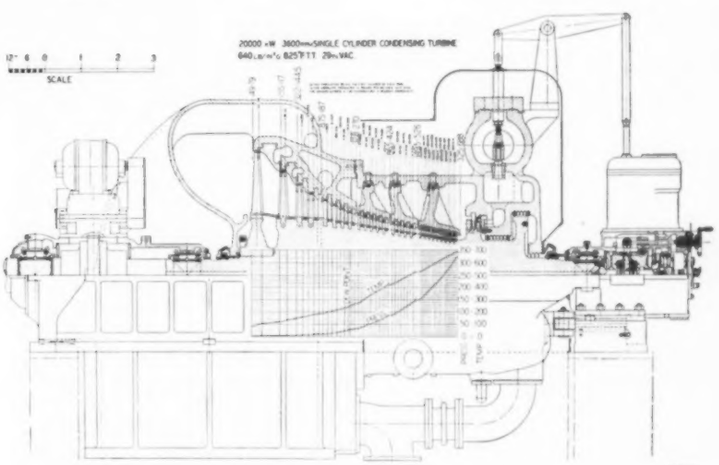


FIGURE 4—Results of "relaxation" tests show a rapid drop in the prevailing stress as temperature increases.

The period from 1928 to 1932 brought single units of tremendous capacity with normal steam conditions. This period provided the start of straight through units, and "tops" of 650 lb. per sq. in., 825 degrees F., provided the intermediate step to the topping turbines which we know today.

In the power plant expansion program from 1934 to 1936, the advances in the art of metallurgy and research on materials provided the possibilities of increasing the inlet temperatures to 925–950 degrees F. Many successful installations have been made and present operating conditions of 1250 lb. per sq. in., 950 degrees F.,

FIGURE 5—A wide variation exists in the normal temperatures prevailing in a single case straight condensing turbine.



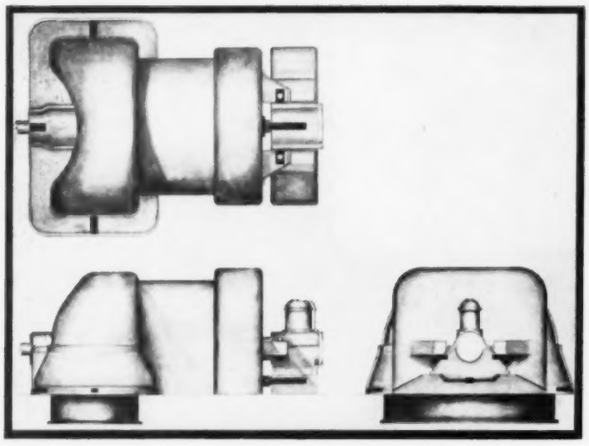


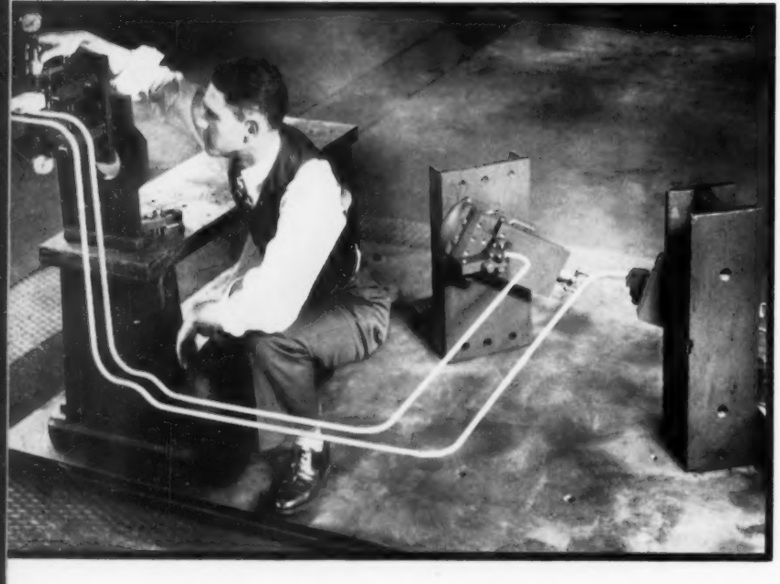
FIGURE 6—Allowance for expansion and provision for perfect alignment are afforded by the means indicated in this drawing.

present no particular problems. There have been experimental plants built for operation at 1000 degrees F. but as yet the application is not widespread.

It is interesting to note that practically all this development centered around the use of 3600 rpm. units and superposed units up to 65,000 kw. and tandem condensing units of 50,000 kw. are under construction.

All of these advances have been made possible by metallurgical development and tedious studies of materials under the influence of stress and high temperatures. The two major problems which confronted the designers were "creep" and "relaxation." "Creep" in general is associated with the growth of material under

FIGURE 8—This scale model of a complicated header piping system is being tested for expansion forces.



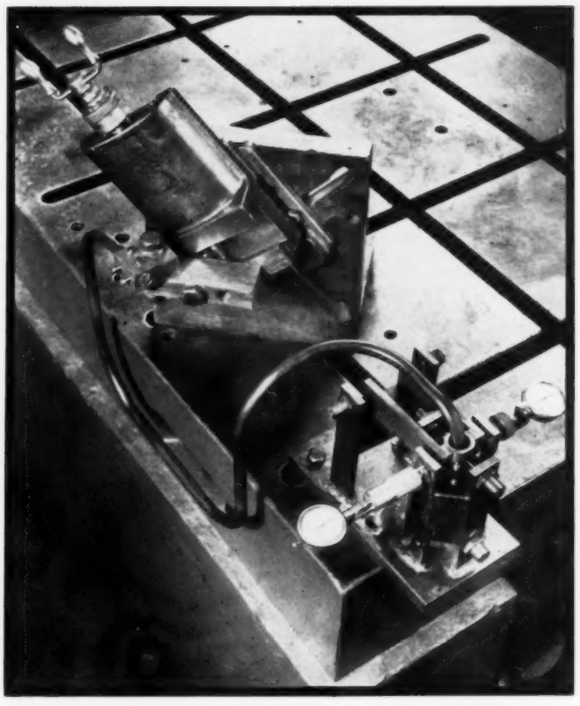


FIGURE 7—Stresses set up in piping systems may be measured from test made on small scale models.

constant stress and "relaxation" with the reduction of the stress due to the growth in the material.

A comprehensive and exhaustive study of "creep" is being carried on in our research laboratory. Such investigations are of long term duration and experiments started in 1934–35 will not be completed until 1944–45. The information is being recorded constantly and the present status of the experimentation plus intelligent extrapolation and mathematical relations give the designer all of the information necessary to produce rational and satisfactory designs.

The chart shown on Figure 2 shows typical information obtained from these tests. It is obvious that permanent deformation under high temperature is influenced tremendously by the attendant stress and in the design of our equipment the stresses are limited to values which provide safe designs.

The phenomena of "relaxation" comes up in conjunction with the bolting members of joints. In the design of rotating equipment it is necessary to provide for assembling and dismantling, and this necessitates the use of joints which must retain the internal working medium during normal operation of the equipment. These joints must be held together by bolts which under all conditions must exert a force greater than the force resulting from internal pressure.

Figure 3 shows a testing machine for establishing the "relaxation" properties of materials. Stress is established in the material and readings of resultant stress with time and temperature are recorded. Figure 4 shows typical test results from the relaxation testing machine. It is obvious that in the "relaxed" condition the bolt

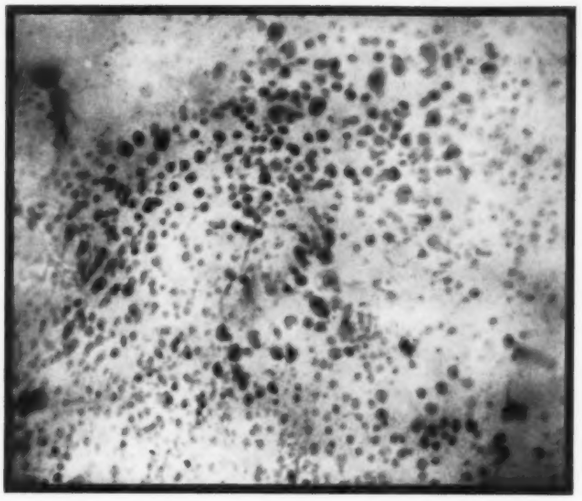


FIGURE 9—Gamma-ray pictures may be used to find casting flaws not apparent from external inspection.

has a surprisingly small percentage of the original established stress on tightening. The designers' problem resolves itself to one of establishing a resultant bolting force with sufficient margin over the internal forces so as to maintain tightness of the joint in question.

Notwithstanding the problems associated with the materials used in construction, the designer is faced with additional problems resulting from temperature influences. Figure 5 shows the normal variation in temperature which exists in a single case straight condensing machine. It can be observed that the temperature variation in a space of approximately eight feet is from 825 to 100 degrees F. The design must be worked out to accommodate this condition and at the same time maintain suitable operating clearances. In addition to this requirement, the unit must be supported in such a manner to allow the expansion to proceed without setting up stresses and distortion. Figure 6 shows the method of allowing for this expansion and at the same time providing for perfect alignment of parts.

The steam piping connection between the turbine and the boiler is also subjected to changes in elongation with resultant stresses which are additional to the turbine expansion. This must be carefully studied and the piping arranged so that the resultant forces and moments are within tolerable limits. Piping systems in general resolve themselves into rather complicated shapes and mathematical solution is very laborious. Complicated systems of this type are studied by scale model tests which provide excellent and simple means with which to determine the resultant forces. Figure 7 shows a close-up of a simple pipe model with the force measuring devices in evidence. Figure 8 shows a model of a complicated header piping system which has been set up for testing. By virtue of the known values of the stresses existing, it is possible to allow greater forces at the turbine which naturally result in a maximum of simplicity in the customer's piping.

FIGURE 10—View of the condition exposed by chipping the section indicated by the Gamma-ray picture of Figure 9.

Welding is used extensively in the construction of high pressure and high temperature piping. Considerable care must be exercised in the welding process and local stress relieving is essential in establishing a satisfactory weld.

The manufacture of the turbine proper involves metallurgy to a tremendous degree. Steel castings must be carefully examined to establish that the material is free from flaws. In order to study the structure of the material "X-ray" and "Gamma-ray" are used exten-

FIGURE 11—In the pull test, the shroud over each turbine blade is subjected to a force four times the normal operating condition.

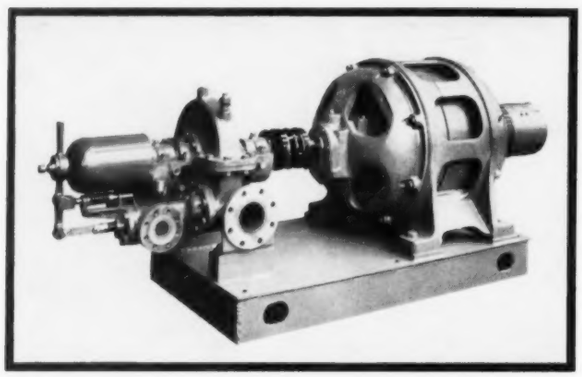


FIGURE 12—View of one of the smallest turbines, being about 14 in. in diameter.

sively. "X-ray" pertains to electrical means of producing penetrating rays by which means pictures of the internal portion of the material can be taken. "Gamma-ray" utilizes the natural radiant energy from radium salt for the same purpose. "X-ray" requires a rather elaborate set-up for operation, while "Gamma-ray" can be carried out with a remarkably simple set-up. The "X-ray" room at our works is completely encased in lead to prevent the escape of the deadly rays. A lead shell encases a radium capsule which is portable and can be used in "Gamma-ray" operations where it would be impracticable to set up the "X-ray" machine.

Figure 9 shows a "Gamma-ray" picture of a section of a turbine cylinder indicating flaws, not apparent from the external observation. Figure 10 shows the condition exposed by chipping the questionable section. Forgings used in the construction of the spindles receive equal painstaking investigation. Test specimens are removed from the forgings and given careful metallurgical analyses. The internal portion of the spindle forging is also carefully investigated by providing an inspection hole through the center of the spindle and observing the bore by optical means.

Alloy steel forgings which need to withstand high temperatures and high stresses receive special attention at the mill. This is known as "heat indication" processing and consists of slowly rotating the rough machined forging in a medium of temperature in excess of the maximum operating temperature to which the spindle will be subjected. This process relieves the internal stresses set up in the alloy steel by the forging process and provides stability in the spindle after final machining.

In designing a steam turbine we are faced with many complex problems resulting from the influence of the operating steam conditions. A few of these will be enumerated in the following discussion.

Differential temperature effect is manifested in admission valve parts such as the diffusers in the steam chest. Due to the flow of hot steam through the diffuser seat, in starting, the diffuser expands at a faster rate than the body of the steam chest. Consequently any press fit at this point would be destroyed and the dynamic force of the flowing jet would tend to lift out the valve seat.

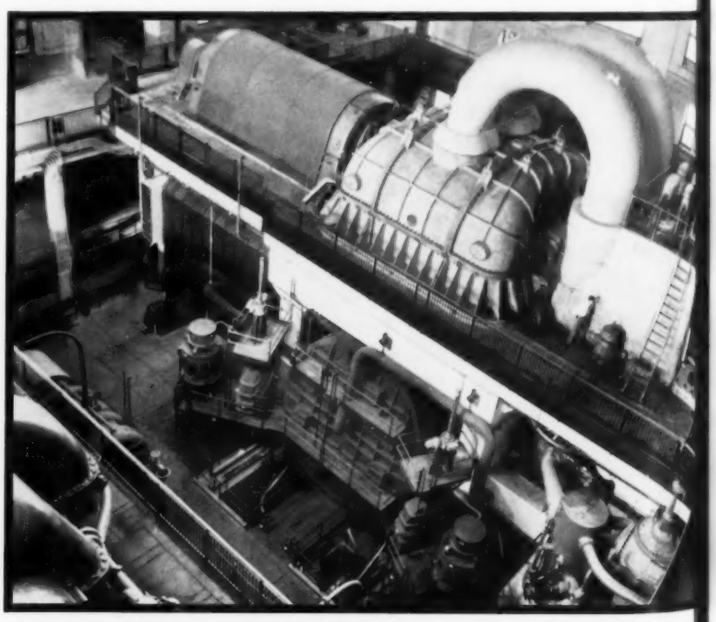
The solution of this problem lies in using a threaded diffuser which is screwed into the steam chest body. It is to be observed that the threaded section has sufficient clearance to accommodate the radial expansion of the valve seat and yet provide for holding it in place. A flexible upper lip is provided on both the valve seat and the steam chest body for the purpose of applying a seal weld. The surface at the bottom of the seat is provided to take any possible side thrust resulting from the steam flow.

Another problem of differential expansion occurs in conjunction with the relation of the rotating to the stationary parts. During the starting phase of operation, the rotating element will grow radially faster than the stationary element and if the differential movement is sufficient, may result in a rub. Also in shutting down, the stationary element will radiate heat at a more rapid rate than the rotating element and exert a clamping action on the rotating element resulting in a rub.

The solution of this problem lies in the proper provision for the stationary element to remain at substantially the same temperature as the rotating element at the time when rapid changes in temperature are taking place. This may be accomplished by the use of separate ring construction for the stationary elements.

The circulation of steam over and around the ring insulates it from the outside cylinder and maintains the stationary ring at a temperature approximately that of the rotating element. The separate rings are mounted by point contact at the joint and held in correct alignment by clearance pins at the top and bottom of the cylinder, thus exerting no expansion forces whatsoever on the outer casing. The blade rings are aligned in the horizontal and axial planes through the center of the machine and maintain that alignment regardless of the change in the shape of the outer casing. The stationary

FIGURE 13—This 165,000 kw. turbine is said to be the largest single shaft turbine in the world.



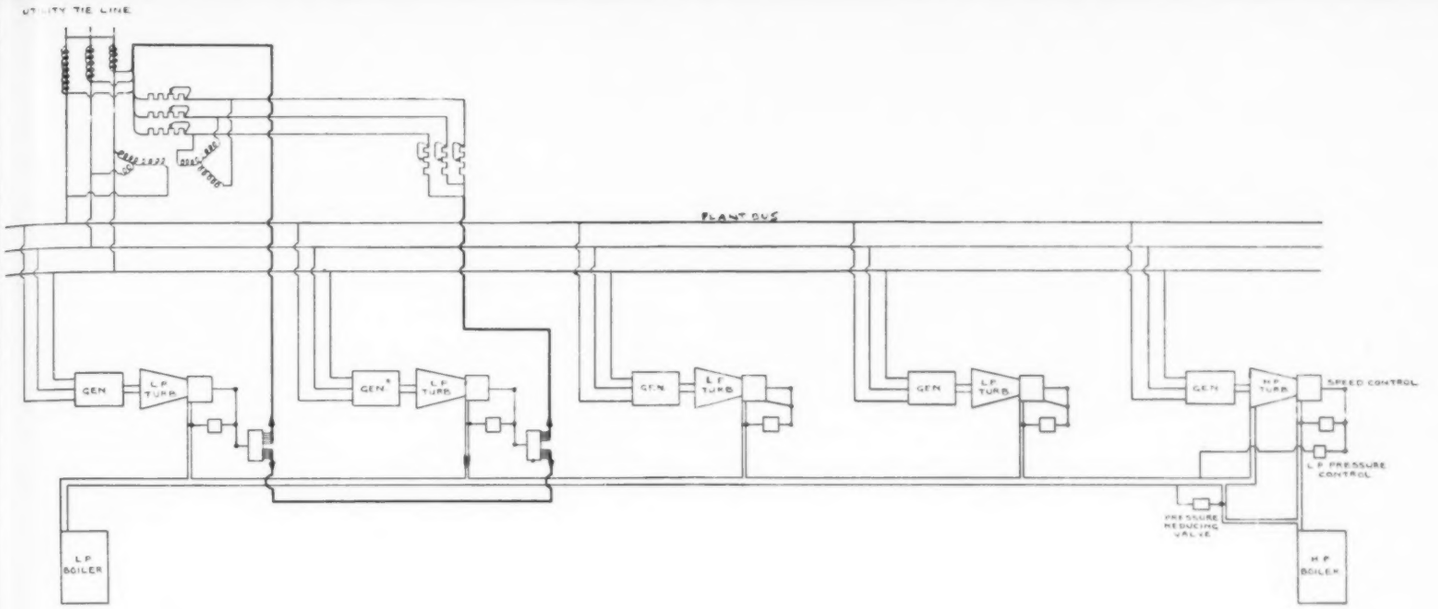


FIGURE 14—Diagram of steam and electric system with control to reduce peak demands from outside tie lines.

blade rings are provided with interstage sealing rings and the differential pressure across the ring maintains the seal.

The mechanical structure of our blading presents another interesting study. The blading is of the radial clearance type. The blades are bonded together in groups by shrouding which is placed over pegs at the ends of the blades and welded securely into place. Any form of radial rub may take place with a minimum of heat generated in the relatively thin radial seals and the heat is readily radiated to the surrounding steam and carried away. The seal strips are rolled into circular shape to conform to the groove and are held in place by rolled and caulked packing pieces.

The welding of the shrouds to the blades is accomplished by a very careful and supervised method and is carefully checked as a part of the manufacturing procedure.

Figure 11 shows the pull test method of checking in which the shroud over each blade is pulled with a force four times the normal force existing in operation. By this method unsound welds are easily detected. After the manufacture, the spindle is assembled in the cylinder and given a test run in the shop. The last operation before dismantling is to run the spindle at twenty per cent above normal speed for a period and this establishes without question the soundness of the blading.

The exhaust end of the condensing type unit presents other interesting problems. Conditions surrounding the low pressure blading have become more critical with the advent of higher inlet pressures and increased turbine efficiency, resulting in higher moisture content in the low pressure zone. With steam conditions of 350 lb. per sq. in., 750 degrees F., 29 in. vacuum, the moisture content in the exhaust is approximately 12 per cent; this is 13 per cent with 650 lb. per sq. in., 825 degrees F., 29 in. vacuum, and with 1250 lb. per sq. in., 900 degrees F., 29 in. vacuum the moisture content is 15 per cent. This increase in the moisture content has been paralleled by an increase in the speed of the blading in establishing larger exhaust annuli to increase rating or to reduce leaving losses. Blade tip speeds have risen from 900 or 1000 ft. per sec. to a maximum of 1260 ft. per sec.

The solution to this problem seems to have been found in the nature of three mechanical and one thermodynamic consideration. First it was recognized that improving the flow conditions through the blading would give a more uniform distribution of the water droplets over the full length of the blade and eliminate water concentration at the outer periphery in the region of high tip speed. This was accomplished by the use of a "constant circulation" blade which makes possible smooth axial flow and eliminates the inefficient conical flow in a tapered direction.

Water catchers have been found to be quite effective in removing the free moisture from the steam. This figure shows the general construction of the water catchers as applied to the low pressure section of a condensing turbine. The water thrown off by the centrifugal action of the steam flow and the rotating blades is caught in the outer cavities and drained off automatically into the condenser.

Also when the length of the blade demands lashing wire in addition to the shrouds, the lashing wire is made in a "tear-drop" or streamlined section. This effectively breaks up the water concentration immediately follow-

(*Please turn to page 32*)

Problems in

STAINLESS STEEL PRODUCTION

By H. A. GROVE, *Metallurgical Dept.*

REPUBLIC STEEL CORP.

MASSILLON, OHIO

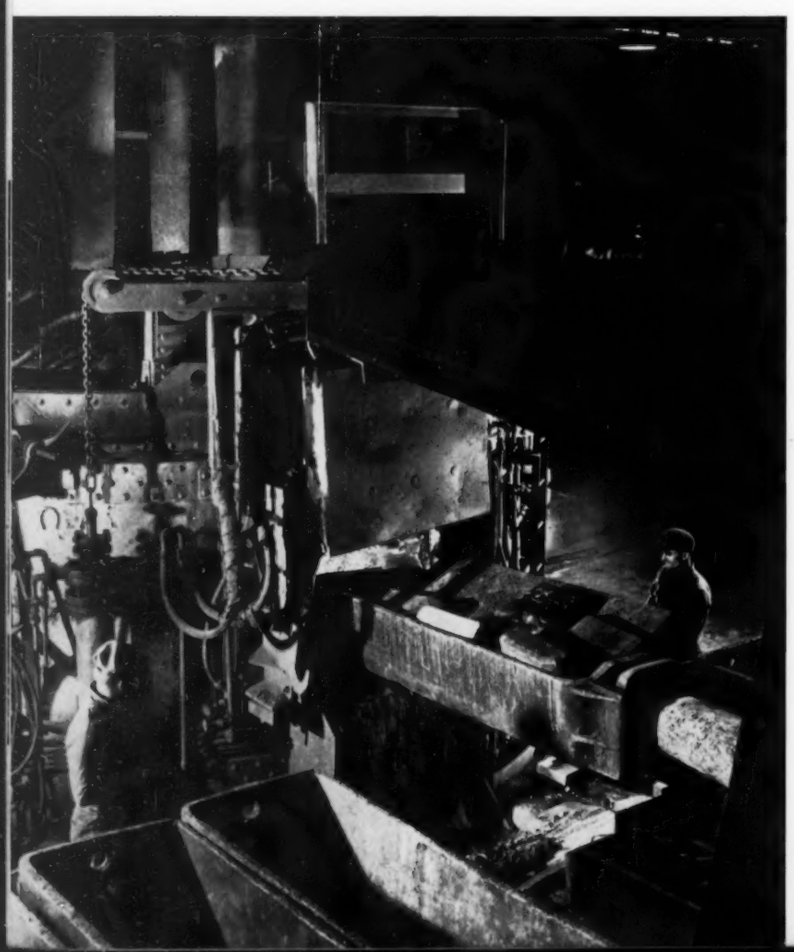
Presented before A. I. S. E. CLEVELAND DISTRICT SECTION

at Canton, Ohio, May 28, 1940

▲ IT has been some twenty-five years since the first alloys, later to be known as stainless steels, were developed. This discovery was made in England and Germany at about the same time. Dr. Brearley developed the straight chrome type in England, while Dr. Strauss in Germany discovered the chrome-nickel grade, which is commonly known today as 18–8.

These two grades grew rapidly into a large number of closely associated alloys, which are known today as stainless steels. There are, according to the last listing by the American Iron and Steel Institute, some thirty-eight different types of stainless steel, each having more or less its own specific use for which it was developed.

While the problems of making, rolling and fabricating of stainless steel have been somewhat solved, or one might say lessened since its first discovery, a large majority of them are still with us and constant care must be employed to insure the highest standard of quality. The development of each new grade has brought its own problems with it. These had to be worked out in order to produce each grade in a satisfactory manner.



Certain precautions must be taken in the manufacture of stainless steel, which is usually produced from basic electric arc furnaces.

Stainless steels may be divided in general into two groups; that is, first, the straight chrome grades and, secondly, the chrome-nickel types. The first group consists of alloys principally of chrome and iron. The amount of chrome, depending upon the grade, will vary from 11 per cent to some 30 per cent. The most popular analysis of this class is the 16 to 18 per cent chrome variety. This grade has been largely used for decorative purposes, especially as moulding trim for automobiles. The lower straight chrome types can be hardened by heat treatment and the hardness obtained depends largely upon the amount of carbon present. These hardenable types are used for turbine blades, cutlery, surgical and dental instruments, bearings, etc. One might say the most commonly used of these hardening types is the 10 to 14 per cent chrome plus various amounts of carbon. The higher chrome, hardenable grades are used where added corrosion resistance is desired. The high chrome types, that is, 25 to 30 per cent chrome, are used for heat resisting alloys. They can be used up to temperatures of 2000 to 2100 degrees, F.

It should be pointed out that the straight chrome types are not as corrosion resistant in all cases as the chrome-nickel steel and that the addition of carbon for increased hardenability will tend to lower the corrosion resistance.

One of the problems which generally presents itself with the fabrication of the straight chrome type is its welding. Welds on this grade have a tendency to be brittle, unless heat treated after welding. This is due to the hardening nature of the alloy. However, this

brittleness can somewhat be overcome by the use of chrome-nickel weld rods or by addition of certain elements, such as aluminum, molybdenum, columbium, etc., to the steel and by keeping the carbon low. These elements in general prevent hardening of these alloys and give a more ductile weld.

Let us turn back a moment to the chrome-nickel grades of the 18–8 variety. These steels in general are austenitic by nature and cannot be hardened by heat treatment as can the straight chrome steels. The chrome-nickel stainless steels are the more corrosion resistant and are the more widely used type where corrosion is the general factor which governs the selection of the proper grade of stainless steel. These grades are in general more ductile than the straight chrome variety and are, therefore, used for deep drawing or severe forming. They may be hardened to high physical properties by cold working. The chrome-nickel grades have one of the widest variety of steel applications known today. They spread from the home cooking utensil and tableware through the dairy and chemical industry, where their corrosion resistance is important, to the transportation field where they are used for their high physical properties in streamlined trains.

To take up some of the problems in the making and rolling of the stainless steels, let us begin with their melting. Stainless steel is melted in most cases in the basic electric furnace. The procedure in melting down and shaping up the bath is somewhat similar to the electric furnace practice employed on other steels. However, certain precautions must be taken; the bath should be free of gases; that is, dead at the time of tapping. Otherwise, the steel will back up or bleed in the ingot form. This killing of the bath and keeping it de-gassed requires a certain amount of technique in slag manipulation. The finishing slag must be either deoxidizing or neutral to slightly deoxidizing. However, in building up such a slag, one cannot employ the usual carbide slag that is used as a deoxidizing slag on alloy or other electric furnace steels.

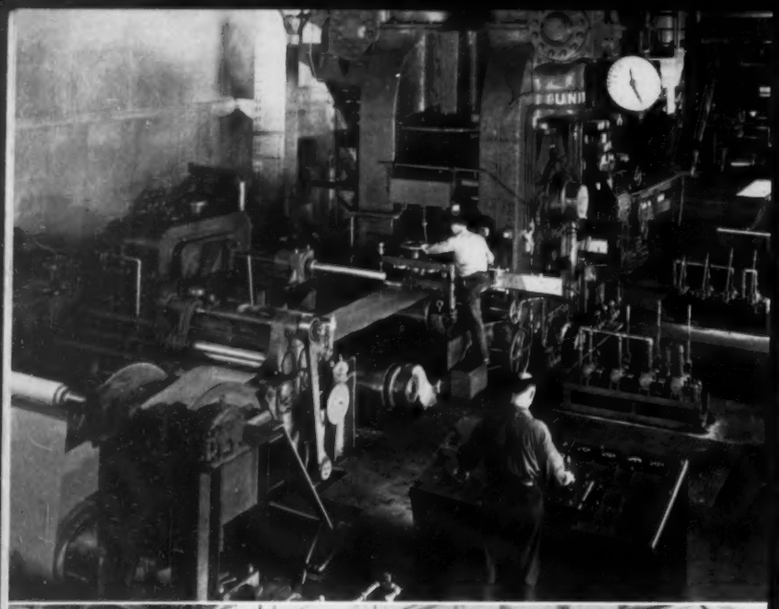
The practice of slab grinding to remove cracks, seams, scabs, slivers, etc., produces a satisfactory surface on the hot-rolled sheet bar or strip.

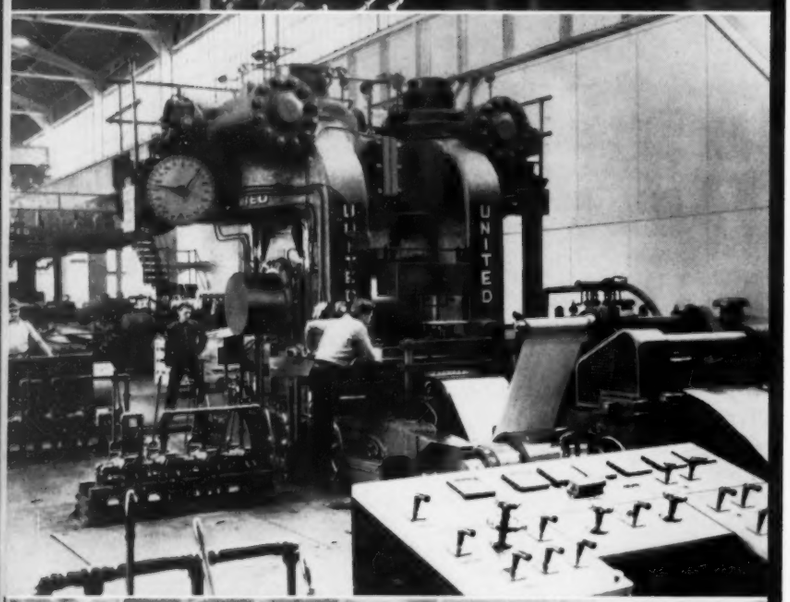
If a carbide type slag was used, a carbon pickup in the bath would be experienced and this is not desirable, especially in the chrome-nickel grades where as low a carbon as possible is desirable.

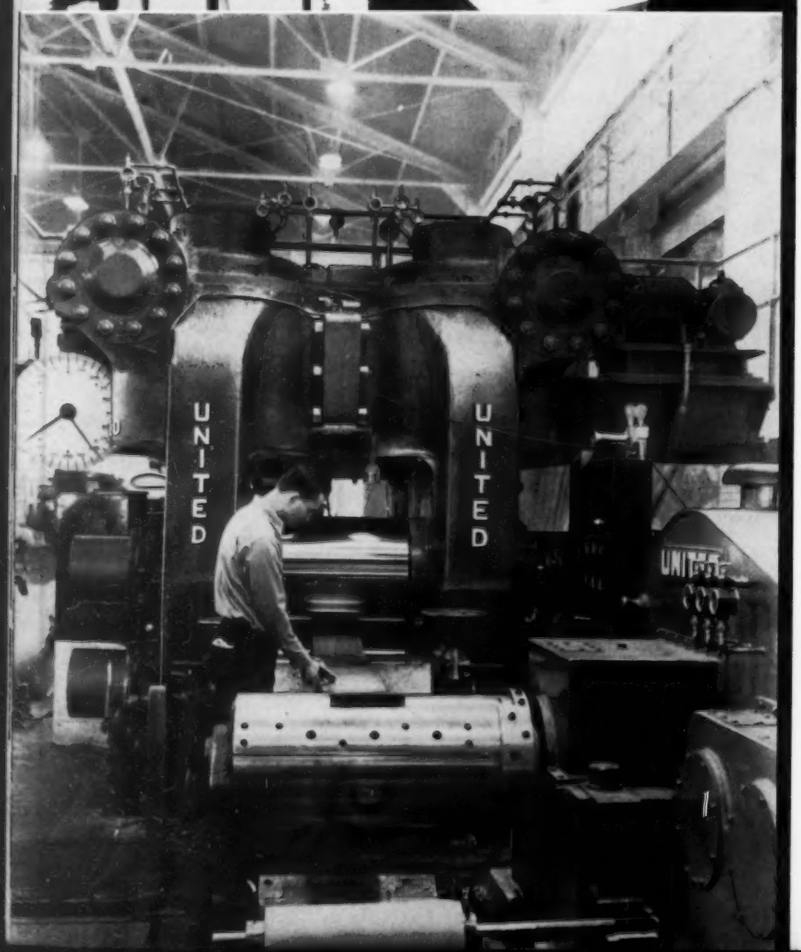
In making alloy additions, the ferro-chrome must always be preheated to prevent chilling of the bath and the tapping temperature should be well up to give the correct fluidity to the metal. The proper tapping and pouring temperature will vary somewhat with the different grades and, in general, it will require the trained eye of an experienced melter to determine properly the fluidity and temperature of his heat.

In breaking down and rolling the ingot more care in

In the hot rolling of stainless strip, heating and rolling temperatures which vary for the different grades, must be carefully controlled.







general must be taken with certain of the chrome-nickel grades. These steels have a tendency to develop a duplex structure of austenite and delta iron. This latter constituent at elevated temperatures is not as ductile as austenite and must be broken up by soaking at the proper temperature. The formation of the delta iron phase is in most cases caused by an improper chrome-nickel balance. However, the addition of other alloys, such as molybdenum, titanium, columbium, etc., will also affect the presence of the delta phase.

The hot rolling of these more tender stainless grades is always a problem, especially the 18–8 molybdenum type, the high chrome (25–30 per cent) grade and the high chrome-nickel grades. These types are very tender in the ingot form and have a tendency to check or crack. It is sometimes necessary to forge or press the ingots for the initial breakdown. Another practice used where forging is impossible and it is necessary to roll is to sadden the ingot several times during this first rolling operation. By saddening is meant the reheating of the ingot between small reductions by rolling. This latter practice is favored over pressing by certain producers, as they feel that a better and more uniform slab is obtained. However, this is an arbitrary question and depends somewhat on the mill and its equipment.

Manganese, when used as an alloy addition between 1 and 2 per cent, greatly improves hot ductility and aids in the initial breakdown and hot working of the ingot. This element has not been found to be detrimental so far as corrosion resistance and other properties are concerned. Therefore, its upper limits have in most

The cold rolling of stainless strip is a succession of alternate reductions and annealing, which must be carefully controlled to obtain the desired finished characteristics.

specifications been raised in the last year or so to 2 per cent maximum.

Following the hot rolling of the ingot, the resultant slabs or blooms of the straight chrome grades are usually buried or pit annealed to prevent cracking, since this grade is of the hardenable type. The cracking may take place at the time of cooling or it may not occur until later when the slab is being ground. Chrome-nickel grades of the austenitic types in general will not crack during the cooling or grinding of the slabs.

Before hot rolling to either sheet bar or hot rolled strip, slabs must be ground all over to remove surface defects, such as cracks, seams, scabs and slivers. The grinding of the slab has been found to be more economical than grinding the finished sheet bar or hot band. This practice of slab grinding produces a very satisfactory hot rolled surface, which only needs to be cleaned and spot ground in the case of sheet bar before additional rolling is done. Hot bands for strip steel are cold rolled without additional spot grinding, being merely pickled to remove hot roll scale.

In hot rolling sheets, slabs are broken down to sheet bar on a jobbing mill. This mill consists of a roughing and finishing stand of rolls. Little trouble is encountered in hot rolling slabs if a good uniform slab has been received and the mills are in proper shape. However,

the rolls must be in good condition, as well as the guides or shoe plates on both the entering and exit side of the rolls. Otherwise, the plate will be gouged or scratched, causing slivers in the finished bar.

Sheet bar, which is usually ¼ to ⅜ in. in thickness, is grit blasted, carefully inspected and spot ground to remove all surface defects before hot rolling to the finished sheet. Hot rolled sheets are generally finished about 25 to 30 per cent heavy to gauge to allow for sufficient cold rolling to improve the surface.

After hot rolling and before cold rolling, sheets are annealed—straight chrome grades between 1400 to 1500 degrees F. and chrome-nickel grades generally well over 1900 degrees F. Care is taken in annealing to obtain a tight scale rather than a loose, flaky scale, as the former produces a more uniform finish on the sheet. Chrome-nickel grades must be cooled sufficiently fast from the annealing temperature to prevent carbide precipitation. To accomplish this in heavy gauge sheet and plate, it is necessary in most cases to quench in water.

Sheets are pickled before cold rolling. The pickling is generally done in about 10 per cent sulphuric acid or sulphuric-muriatic acid followed by a short pickle in nitric-hydrofluoric acid. This produces a white, uniform finish. Some producers prefer to use a muriatic acid pickle on the straight chrome grades, reserving the nitric-hydrofluoric pickle for the chrome-nickel types. In any case the chief objective of pickling is to remove all scale before cold rolling. Otherwise the scale would be rolled in, leaving a rough, objectionable looking surface on the finished sheet. Following pickling, the sheets are washed in water, scrubbed and dried.

The usual care must be taken in cold rolling so as not to produce flopper or pinch marks or other mill marks that might result from a bruised or poorly finished roll.

The sheets after cold rolling are again annealed, pickled and skin passed to the desired finish after which they are either roller leveled or stretched or both for flatness.

Polished sheets are generally furnished in four finishes; namely, No. 4, 6, 7 and 8. These are as follows: A No. 4 finish is a ground finish sheet with the last grinding operation being done with 120 grit.

The No. 6 finish is a tampico or satin finish, is obtained by dulling a No. 4 finish with a brushing operation, using pumice and kerosene as an abrasive.

The No. 7 finish is a buffed No. 4, which has a high lustre but does not have all the grinding lines removed.

The No. 8 finish is the mirror finish and is made by finish grinding with a fine abrasive and then buffing to remove all grinding lines and bring up a high lustre finish.

The preparation of a polished sheet also has its problems. The first is the original sheet surface. A good surface here will decrease the amount of rough grinding, but here again a question arises of whether rolling or grinding to a good surface is the more economical.

Pits and slivers must be spot ground out in the early stages. However, one is occasionally missed or does not show up until the final inspection. This causes a refinishing of the sheet, and added expense.

Flatness is another problem. Since grinding and polishing both produce heat, they have a tendency to warp and buckle the sheet. This can usually be taken care of by stretching and grinding the reverse side of the sheet to flatten it.

Light gauge sheets are very difficult to polish and obtain a flat, smooth appearing sheet. The sheets, no matter how much care is taken, have a tendency to pound in the grinding and polishing and do not have a smooth looking surface if viewed from certain angles.

To return to the rolling of strip steel, here again the slab must be well ground before hot rolling. The heating and rolling temperatures must also be carefully controlled, because there is no intermediate stage as in the rolling of sheet. Too high a rolling or soaking temperature will produce an open surface condition appearance; this rough surface when once obtained is hard to heal and may even carry through to a finishing gauge as light as .010 in.

The various stainless grades all hot roll differently, some giving more trouble than others. The most difficulty is experienced with those types which give trouble in the original ingot rolling.

The chief care in the rolling of strip is the obtaining and keeping, during rolling, of a good surface. The proper amount of cold rolling between anneals helps in obtaining a good finish. Reannealing strip without cold working has a tendency to produce a loose, flaky scale, causing an open or slightly rough appearing surface. This in most cases is very objectionable.

Besides the problem of finish in cold rolling strip, there are the usual ones which present themselves in the working of cold strip, namely, flatness, gauge, etc.

The chrome-nickel grades work-harden very fast; therefore, in rolling this analysis, the heaviest reductions must be taken first. Otherwise, difficulty may be encountered with obtaining the desired amount of total reduction in the least number of passes.

Strip steel is usually given about 75 per cent cold reduction in the first cold reducing stage, after which it is annealed, pickled and cold reduced at 50 per cent

Stainless sheets may be ground to a number of surface finishes on machines similar to the one shown here.

Very narrow widths of strip may be produced by the slitting of wide strip, accomplished on machines as here shown.

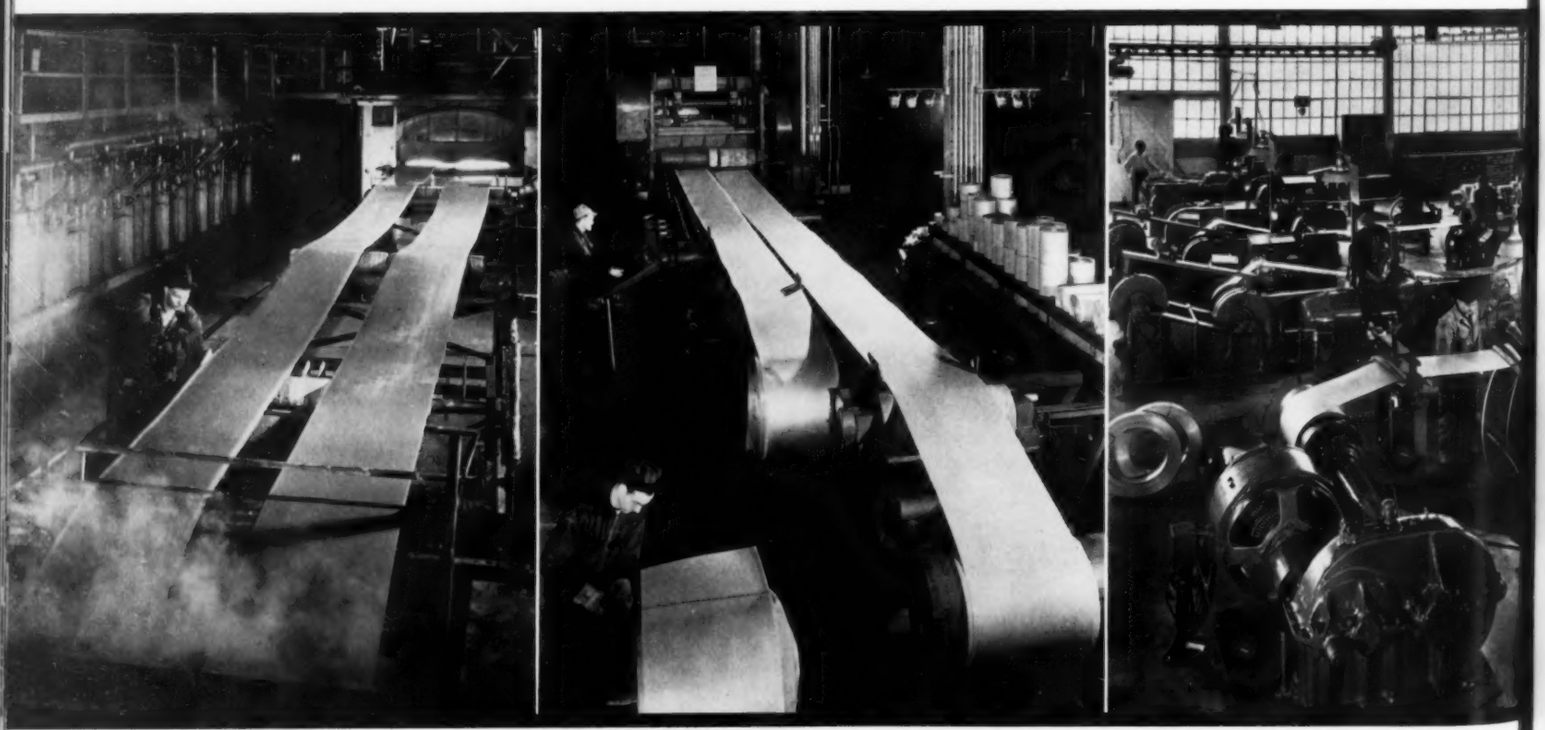
reductions between anneals to the finished gauge. Here it is given its final anneal, pickle and skin pass, after which it is slit to width, inspected and is ready for shipment.

High tensile strip, 150,000 lb. and higher, such as is used in the railway, airplane and allied industries, is produced by a carefully controlled final amount of cold reduction. The light gauge, wide width, high tensile strip has the most problems because here only a .001 in. or so in gauge variation may amount to quite a large per cent of the finish gauge and thereby throw off the final physical properties. As an illustration, .002 in. variation of .010 in. gauge is 20 per cent, whereas at .125 in. gauge it is less than 2 per cent. This gauge problem is important because the analyses used for high tensile work are very sensitive to cold working and a reduction of 5 per cent may cause as much as 10,000 lb. per sq. in. variation in the mechanical properties.

Flatness is also a problem with light gauge, high tensile strip, especially in the wider widths. It is practically impossible to obtain an absolutely flat strip. All coils will either show some wave or rippled condition. This does not seem to be objectionable, however, to most users, as they appear to be able to take care of these slight waves or ripples in their fabricating practice.

The problems of the fabricator of stainless steel is another story and we will not attempt to go into them here. However, stainless today is rapidly entering into many new fields. It has already invaded the dairy and canning industries, the chemical and allied subjects, the transportation field, which includes automotive, railroad and aircraft. It is used both in heating and refrigeration. Its decorative applications are too numerous to mention. The field of stainless is increasing daily, not only because of its resistance to corrosion and its attractive appearance, but also because of its mechanical properties which combine both high tensile strength with good ductility and toughness.

The finishing of stainless strip requires a great amount of equipment, including annealing furnaces, pickling units, slitters, etc.



STAINLESS *Clad* PLATE

By R. E. KINKEAD

Consulting Engineer

CLEVELAND, OHIO

Presented before A. I. S. E. CLEVELAND DISTRICT SECTION, Canton, Ohio, May 23, 1940

▲ EXPERIENCE in making and rolling solid stainless steel has made possible the making and rolling of stainless—carbon steel composites. While there is no record of its, everyone involved in the early manufacture and rolling of stainless steel must have thought of the desirability of making a steel that would be stainless on its surface and ordinary steel between the surfaces. Certainly every man who has nursed solid stainless ingots through a heating furnace preparatory to rolling, has wished that there was some way of reducing the time required for heating and generally decreasing the temperamental behavior of the metal during the early rolling operations. An ingot coated on two sides with stainless steel can be heated almost as rapidly as a carbon steel ingot, since there are two sides remaining which have relatively high thermal conductivity.

The rolling of carbon steel having a stainless surface is not nearly so difficult as might be expected. While there is, of course, dendritic structure in the stainless steel, it is perfectly obvious that the crystals are not as large as those encountered in the solid stainless ingot. In spite of the fact that thin coverings can not be ground with a large swing grinder, little difficulty is encountered due to the cracking of the stainless steel during the initial rolling operations.

There are two processes known which each produce a perfect bond between the stainless surface and the carbon steel. One of them is similar to the metallic arc welding process as illustrated by the Unionmelt process and the other is a carbon arc process in which the author is interested.

In the carbon arc process, the metal of the surface of the ingot or slab being treated is used for the ferrous component of the alloy surface. Either all of the ferrous component or part of it may be obtained from the surface. In the case of the chromium nickel alloys, ferrochrome and pure nickel shot are spread uniformly on the surface to be alloyed. The surface is then covered with a layer of furnace slag. Carbon arcs are then caused to transverse the surface as they progress longitudinally. The purpose of this motion is to get uniform penetration into the surface and therefore uniform chemical analysis of the alloy. The carbon arcs operate through the slag as distinguished from operating under a blanket of flux, as is the case in the Unionmelt and modified processes. The purpose of the furnace slag is to exclude air from contact with molten metal and thus eliminate or reduce oxidation.

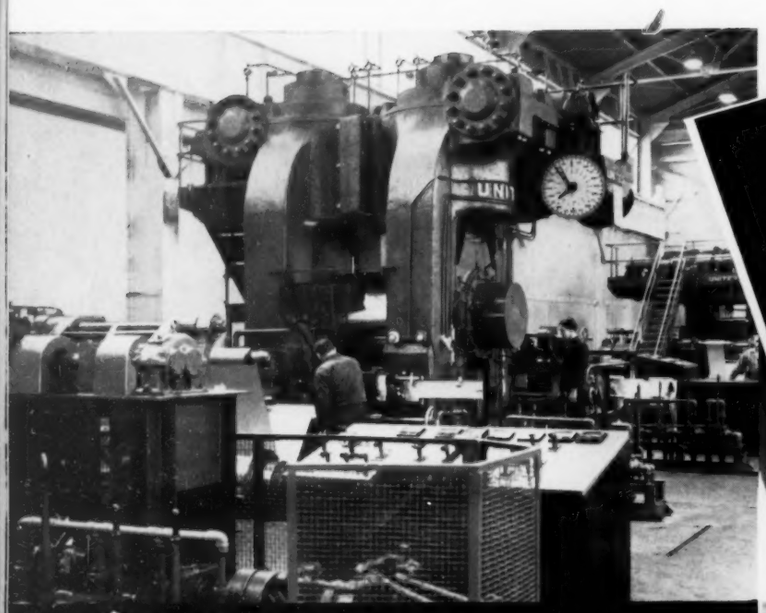
The economics of these methods of making stainless steel composites is obscured for the moment by the fact that they are new and much is to be learned about their economical production. But at least let it be said that we know what we are shooting for and that the goal is within sight; namely, the production of a carbon steel having a very thin coating of stainless steel on both sides to sell at a price to compete with tin plate and galvanized. It appears that this field is not in any sense competitive with the field for solid stainless steel.

Putting thirty to forty lb. of tin on a ton of steel makes tin plate. At 50 cents a lb. for the tin, this means a tin cost of from $15 to $20 per ton. The mark-up due to the tin is about $30 a ton, so that the difference between the cost of the tin and the cost of the mark-up represents the cost of putting on the tin and the profit. For some queer reason, the steel industry has difficulty imagining a coating of stainless steel on carbon steel as thin as the coating of tin on tin plate. Yet, in our early and crude experiments with no particular effort whatever to get a very thin coating, we were able to roll steel with a coating of three ten-thousandths of an inch of stainless steel. It is not difficult to believe that tin plate thickness of stainless steel on carbon steel is possible and will give a product with which we may do things that can not now be done with tin plate or galvanized. Eighteen and eight stainless steel is superior to tin in practically every respect.

The usual reaction to this on the part of the steel industry is the question as to how you are going to condition such a surface. The answer to that question is that you are not going to. In the first place the handling of thinly alloyed surfaces is not so difficult as it would seem. In the second place, the value of the material is such that it won't pay to condition it even if it were possible. Another difficulty arises from the fact that many of the makers of solid stainless steel are mentally conditioned to the extent that they are virtually in the jewelry business. Making and selling a product that sells from $600 to $2000 a ton and earning profits doing it, is a very different job from that of a steel mill. Making steel for soda fountains and decorative purposes and high-priced cooking utensils is one thing and making steel that can be sold for use in hog troughs and barn roofing is something else.

In passing, it is significant that several million tons per year are required of the low priced product against a present hundred and fifty thousand tons of the jewelry classification.

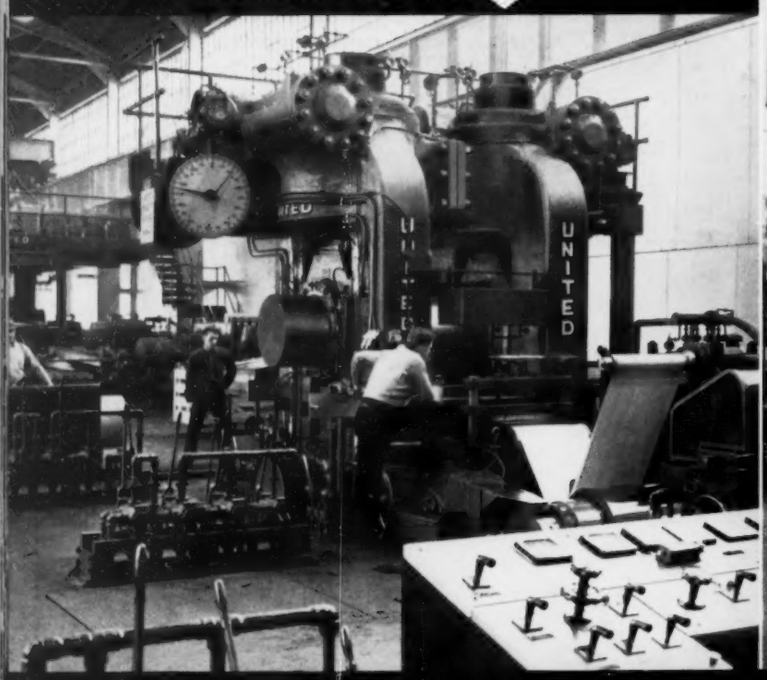
ROLLING STAINLESS



34" REVERSING COLD MILL

2-HIGH TEMPER PASS MILL

28" REVERSING COLD MILL



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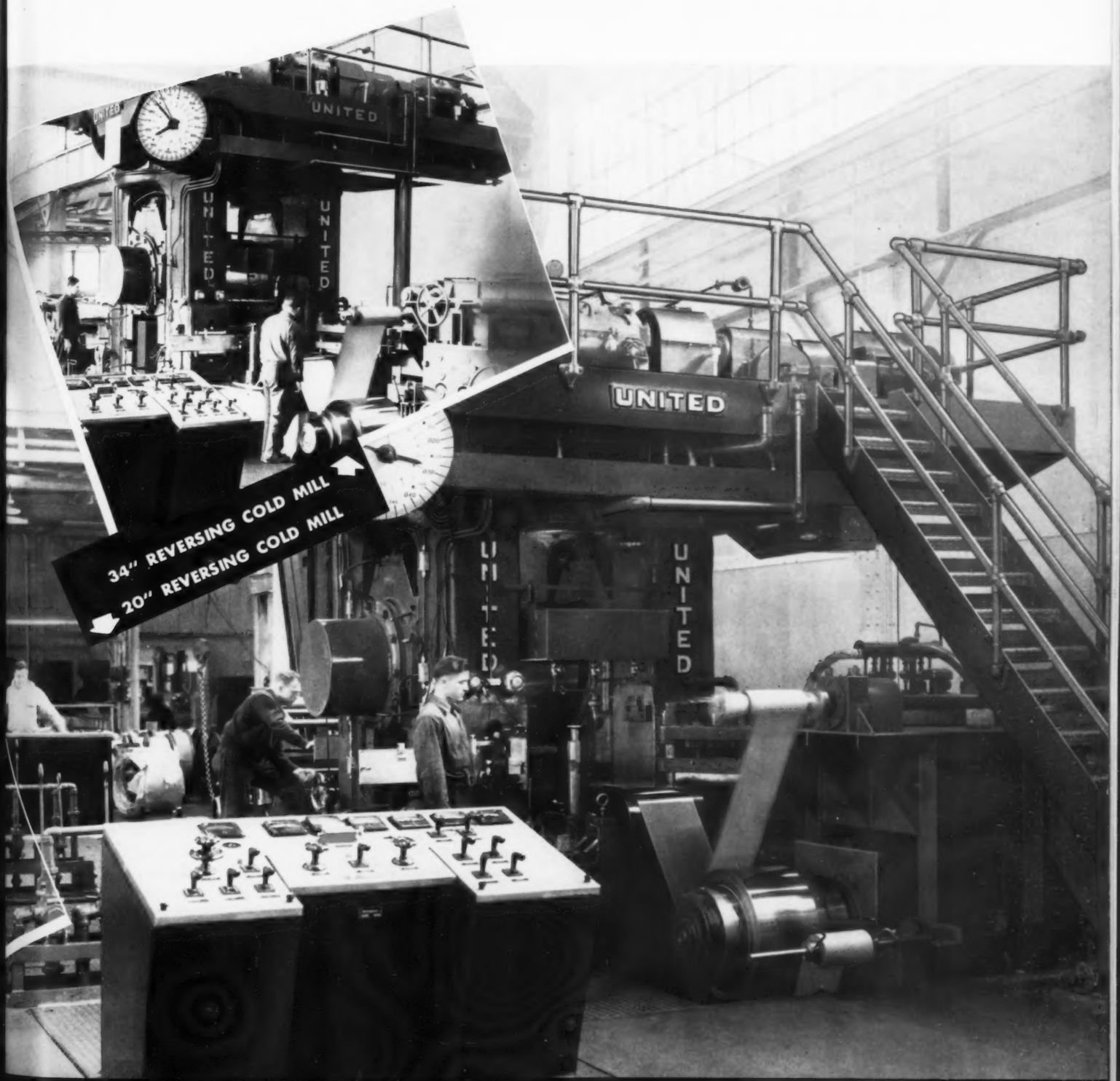

VORKS, L' MONTREAL, P. Q. · UNITED INTERNATIONAL, S. A. PARIS, FRANCE · SHIBAURA-UNITED ENGINEERING CO., TOKYO, JAPAN

PHILADELPHIA DISTRICT SECTION
ASSOCIATION OF IRON AND STEEL ENGINEERS

Saturday, December 7, 1940

INSPECTION TRIP

to

BETHLEHEM STEEL COMPANY

CONTINUOUS BUTT WELD MILLS AND ROD MILLS

Maryland Plant, Sparrows Point, Maryland

Buses Leave Belvedere Hotel, Baltimore, Md., at 1:00 P.M.

DINNER AND MEETING

Belvedere Hotel, Baltimore, Maryland

DINNER—6:00 P.M. **MEETING—7:00 P.M.**

"SOME OPERATING PROBLEMS OF CONTINUOUS PIPE MILLS"
By L. J. HESS, *Superintendent, Pipe Mills, Bethlehem Steel Company, Sparrows Point, Maryland*

"OBSERVATIONS ON DESIGN AND OPERATIONS OF ROD MILLS"
By LOUIS MOSES, *Superintendent, Rail Mill and Roll Department, Bethlehem Steel Company, Sparrows Point, Maryland*

Trends in Steel Mill Power Generation

(*Continued from page 25*)

ing the lashing wire and eliminates excessive cutting of the outlet edge of the blades.

Finally the rotating blade is protected by shields hooked around the inlet edges and silver soldered to the blade section. Stellite has been found to be most suitable and effective material for erosion protection and stellite shields have been used successfully for a number of years.

Operating experience with the new type of blading has been extremely gratifying and the erosion in the high speed blading operating with high moisture content has been reduced to an unbelievable minimum.

Figure 12 shows one of the smallest turbines, being about 14 in. in diameter, while Figure 13 shows the largest single shaft turbine in the world. This latter unit is rated at 165,000 kw. and the size comparison can be made by observing the man by the side of the machine.

We have just completed a very interesting control study and application in conjunction with a large steel mill. The problem has been to anticipate and reduce the peak power demands from the outside tie lines on a system of basic rolling mill load characteristics.

Figure 14 shows a typical diagram of the steam and electric system. The plant consists of four old low pressure condensing turbines, one new superposed high pressure turbine and new high pressure boilers. The new high pressure turbine furnishes steam to the low pressure header in response to the demand of the low pressure units. This is accomplished by an exhaust pressure regulator which controls the pressure at the exhaust of the high pressure machine by acting on its governing valves. The high pressure boilers being equipped with rapid control mechanism can respond quickly to system demands and provide a very flexible set-up. The governing mechanism of two of the old low pressure units are electrically connected to the power tie line through a load anticipator and rapid changes in load are absorbed by these units. The use of this mechanism aids tremendously in relieving tie line swings and provides an excellent means of absorbing the power fluctuation incidental to steel mill demands.

Symposium
. . . on Equipment for . . .
MILL CRANE BRIDGE DRIVES
Presented before A. I. S. E. Pittsburgh District Section, April 22, 1940

Symposium

on Equipment for Mill Crane Bridge Drives

AUTHORS

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J. C. COX, Automotive Engineering Division, Wagner Electric Corporation, St. Louis, Missouri

INTRODUCTION

By

F. W. CRAMER

▲ THE Association of Iron and Steel Engineers has been interested in specifications for heavy duty overhead traveling cranes for many years. In 1910 the Standards Committee presented the first set of crane specifications that were adopted by the Association. These were revised in 1916, again in 1921 and in 1929.

During the past several years, many of the steel companies have been questioning the total installed cost of modern steel mill cranes and asking if we were taking advantage of improved design and better steels that are now on the market. In order to answer these criticisms, the Board of Directors of the Association decided that the 1929 Standard Crane Specifications should be reviewed and brought up to date and a committee was appointed to do this work.

The first meeting of the committee was held in January, 1938, which was attended by representatives of all the major steel companies. Later the committee was expanded to include representatives of the crane manufacturers, and investigations inaugurated in several of the sub-topics of the specifications.

One of these subjects was the relation of the maximum wheel loads for bridge track wheels in relation to wheel diameters and size of runway rails. Another concerns gear and pinion specifications in order that wear, life and factor of safety can be correlated. A third problem deals with the proper application of antifriction bearings to bridge track wheels. In a recent preliminary report this subcommittee suggested that mill cranes should be divided into several classes, based on type of service, before definite factors be applied to antifriction-bearing specifications.

Perhaps the most unusual work of the committee is the research program sponsored wholly by the Association and being carried on in the Fritz Laboratory at Lehigh University. This program concerns the analysis of box girder design as applied to cranes. This work is

being done by I. E. Madsen under the supervision of the University and with the cooperation of the active committee and the crane builders. Many interesting facts have already been developed, and the program has recently been extended for another year. At its completion we feel that the results will be a worthwhile contribution from the Association to the steel industry.

Another of these subjects that is of universal interest to both the steel industry and the crane builders is the one to be discussed this evening—crane bridge drives. Many of us feel that most bridge drives are over-motored, that little or no attention has been paid to gear ratios, to the advantages of antifriction bearings, to taper tread wheels, and to a real analysis of the work required by the bridge. There are many installations where over-motoring results in excessive bridge speeds perhaps 1000 ft. per min., and when this occurs it presents difficulties in stopping the bridge and gets beyond the place where bumpers and rail stops are effective. We also have the problem of whether one, two, or four motors are required for bridge drives and how many wheels should be driven.

These few factors are mentioned to show how involved the problem of crane bridge drives has become, and in order that its many phases could be brought to your attention we have asked thirteen men representing both crane and electrical equipment manufacturers to present some condensed facts on this problem.

RELATION OF SPEED TO PRODUCTION REQUIREMENTS

By

R. J. HARRY

▲ THE importance of bridge speed of cranes varies with the application. Where cranes are located on short runways a high rate of acceleration, which can be obtained by relatively small motors and large gear ratios, is most desirable as the crane movement must, of necessity, be short and the bridge rarely reaches top speed.

Where the runways are long and the crane is required to make long movements, less emphasis must be placed on the rate of acceleration and more emphasis on the top speed, and relatively large motors are required and smaller gear ratios.

In many cases the length of the runway does not govern the speed of the bridge. This is especially true when most of the work is done by the hoist as, for instance, in the case of a scrap drop crane. In this application the speed of the hoist is of prime importance and the speed of the bridge and trolley of secondary importance. This can probably best be illustrated by an example of a crane which we built several years ago for one of the steel plants in the Chicago district. This scrap drop crane was equipped with two 200 hp. motors on the hoist, the hoist having a capacity of 25 tons and a hoisting speed, at full load, of 150 ft. per min. The bridge of this crane was equipped with an 80 hp. motor with a speed of 250 ft. per min. and the trolley with a 25 hp. motor at 132 ft. per min.

It can readily be seen that in a crane of this type, a high speed bridge would be of no particular advantage, in fact it might work to a disadvantage inasmuch as both the bridge and trolley motions are used chiefly to spot the load and this function can be accomplished better with a small motor and a high gear ratio on the bridge than would be the case with a large motor and small gear ratio. On a scrap drop crane the only time the bridge motion and trolley motion would be used to any extent would be in loading and unloading cars and as a consequence, for this type of application a moderate speed on the bridge is entirely satisfactory.

In the case of warehouse, or shipping cranes, greater emphasis would have to be placed on the speed of the bridge motion and as a result cranes of this type are usually designed with relatively high bridge and trolley speeds and relatively slow speed on the hoist motion, as in most cases the lift of the crane is rather short and the movements required of bridge and trolley are relatively long. On cranes of this type rated bridge speeds of 600 ft. per min. are not uncommon.

One of the most severe applications for cranes in steel mill service is the open hearth stock yard crane. In most applications these cranes have relatively short lifts and are frequently called upon to cover considerable yard areas. In this particular application speed is a prime requisite on all motions as most of the loads are handled by means of lifting magnets, requiring no attendant, and in order to keep up production the time required to pick up a load from the car and deposit it in the charging box should be kept to a minimum.

In most open hearth stock yards the charging boxes are usually placed parallel to the cars of scrap so as to reduce the amount of bridge motion to a minimum. Where this arrangement can be made the speed of the hoist and trolley are of greater importance than that of the bridge, but in many cases the relative location of the cars of scrap and empty boxes cannot be conveniently arranged and as a consequence the bridge is called upon for greater use and the bridge speed becomes of greater importance. We find in checking over our records that stock cranes are usually specified with speeds of 400 to 550 ft. per min. depending upon the service required.

There are many steel mill applications where it is necessary for a floor attendant to attach and detach loads, such as, for instance in mill machine shops or certain types of yard cranes, and in such applications it would obviously be necessary to coordinate the speed of the bridge with the walking speed of a man. Under such conditions an ideal bridge speed would be 350 ft. per min. Any greater speed than this would simply mean the crane would be detained at its destination waiting for the floor attendant to attach or detach the load.

Another application of steel mill cranes is the mill repair crane, sometimes called the roll crane. These cranes are usually of relatively high capacity and on account of their infrequent operation are relatively slow operating. These cranes are usually equipped with relatively slow, large capacity main hoists; small capacity, high speed, auxiliary hoists; and relatively slow

trolley and bridge motions. For cranes of this type bridge speeds of 200 to 250 ft. per min. are usually satisfactory, the bridge being powered with relatively small motors and relatively high gear ratios so that the heavy loads can be spotted with accuracy. Steel mill motor house and roll shop cranes come under this same category on account of the relatively heavy loads to be handled and the accuracy with which these loads must be spotted.

In discussing the bridge speed of cranes, we usually refer to the speed rating given by the crane builder and this is usually based on the full load speed of the motor. On relatively short movements, the rated speed of the crane will be attained in the course of a few seconds and top speed seldom attained. However, where the cranes operate on long runways and the bridge movement becomes relatively long, in many cases speeds up to 2½ times the rated speeds are attained. This is especially true where series d-c. motors are used, and it is not at all uncommon for cranes that have a normal rating of 500 ft. per min. to actually attain a free-running speed of 1000 ft. per min. when the runway is long.

Much misunderstanding has resulted from the so-called rated speed of bridge motion of cranes, and great care should be exercised in the selection of the proper speed of cranes using alternating current. We consider a direct current crane, with a nominal rating of 600 ft. per min. a high speed crane. If this crane is liberally motored, as most steel mill cranes are, and the runway in good condition, the top free-running speed will probably be 1200 to 1500 ft. per min. Many engineers ordering alternating current cranes with what they consider high speed bridges, are very much disappointed with the operating characteristics obtained. They know that 600 ft. per min. direct current crane is a high speed crane but fail to take into consideration the fact that it is necessary to specify a speed of approximately 1200 ft. per min. for the alternating current crane to obtain the same operating characteristics.

An engineer, in figuring the size motor required for a specified speed of bridge has to consider several factors, some of which are mechanical efficiency of the drive, rolling friction, flange friction, journal friction, rates of acceleration, gear ratios, lubrication, and condition of runway.

Friction alone determines the top free-running speed of the bridge motion for direct current series motor driven cranes, and since this factor is dependent on so many variables no one is able, with accuracy, to predetermine the top speed of any crane. The total friction is often expressed as rolling friction, or draw bar pull, and is expressed in pounds pull per ton of crane and load.

In checking up some published data on crane bridges equipped with brass bearings I found that the rolling friction per ton varies from 4.55 lb. to 30.9 lb., and in a more recent paper by James A. Jackson he records values for roller bearing cranes ranging from 9.38 lb. to 14.27 lb.

VARIOUS ARRANGEMENTS OF BRIDGE DRIVES

By

H. W. BALL

▲ AMONG the engineering problems which are encountered in connection with crane bridge drives, one of the first to be considered is the determination of the number of wheels necessary to suit the span and capacity of the crane and the number of these wheels which should be driven.

There has been a gradual advance in the capacity, speed and efficiency of cranes along with the growth of industry and especially to keep pace with the great expansion in the steel industry. Various types of bridge drives have been developed to meet various conditions or to secure certain advantages in operation or maintenance.

Typical of the very early type of cranes is one driven by a square shaft which ran the full length of the building and was belt driven from a steam engine. At the end of the shaft was a loose and fixed pulley, the belt being shifted from one to the other pulley to start and stop the shaft. The power from this square shaft was transmitted to the various operating mechanisms of the crane through a series of clutches driven by gears and sliding sleeves. Separate clutches were required for each forward and reverse movement, making six clutches in all. If an auxiliary hoist was required, another set was added.

With this complicated system, developments in crane design were very much limited. However, about the year 1881 the first electric traveling crane was built by our company. This was operated by a single motor driving the longitudinal shaft in the building, which in turn drove the main drive shaft of the crane by means of the sliding sleeve and clutches in what was then the conventional way.

From this first application of the electric motor to a crane it was a natural step to the application of separate motors to each individual drive as in Figure 1. This is the simplest type of modern bridge drive with two of the four trackwheels driven. A single line shaft, sometimes called the squaring shaft, is driven by a single motor placed near the center of the bridge span, with one gear reduction at the motor and one at each end. This is known as the "overhung type," the trackwheel axle gears being outside of the truck.

Figure 2 is quite similar to Figure 1 except that the trackwheel drive gears are mounted on the axles between the trackwheel hubs and the axle bearings. This is known as the "inside drive type" and is considered superior to the design shown in Figure 1 from the standpoint of safety since it is considered possible for the

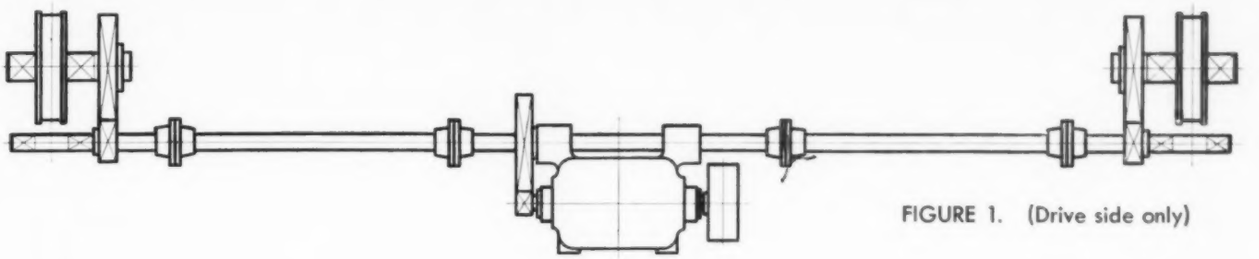


FIGURE 1. (Drive side only)

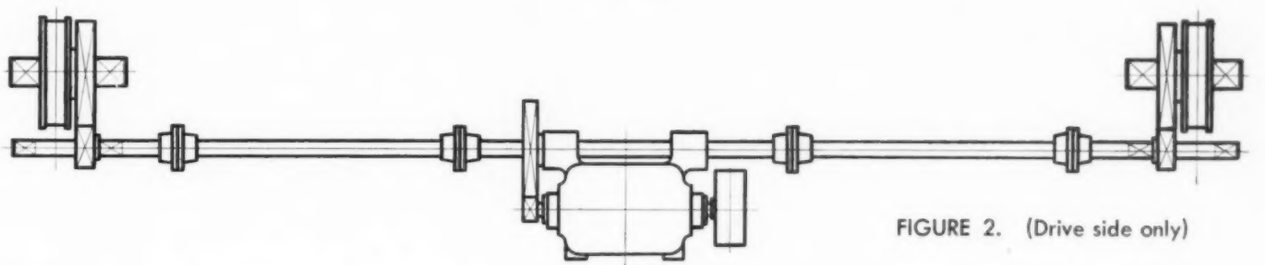


FIGURE 2. (Drive side only)

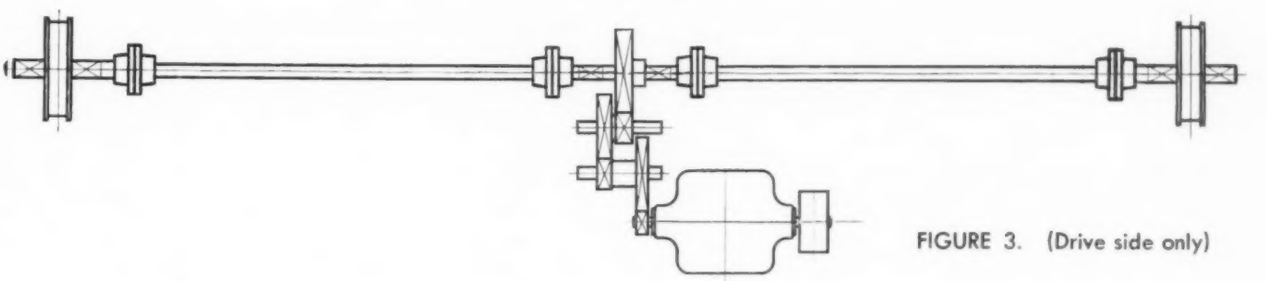


FIGURE 3. (Drive side only)

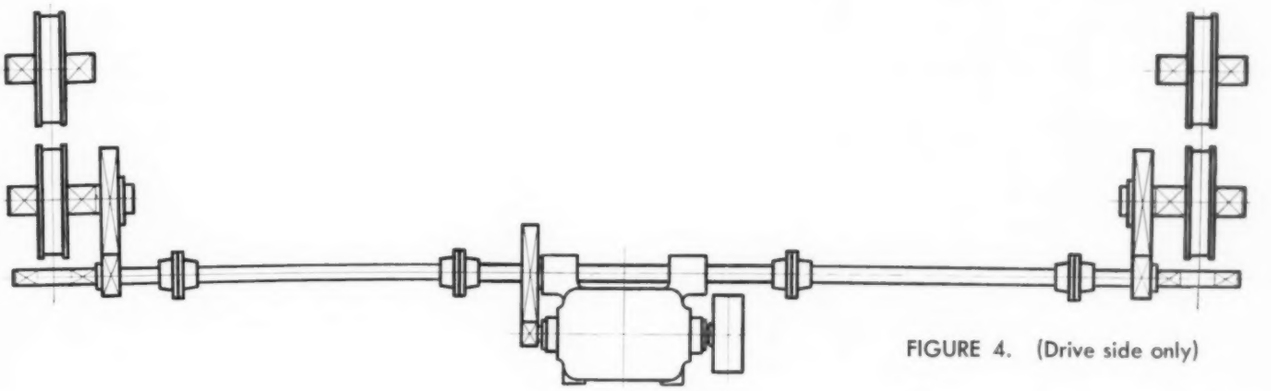


FIGURE 4. (Drive side only)

overhung gear to fall to the floor in case of breakage. Disadvantages of this arrangement are the unequal loading of the two axle bearings or to avoid this greater width of the building in order to provide for a wider truck. With the increasing use of roller bearings the objection to unequal loading has largely become obsolete.

Figure 3 shows another design of modern bridge drive which continues with the single line shaft driving two of the four bridge wheels but eliminates the question of inside or outside drive at the trucks by housing all the gears in a single gear box near the center of the span. This design follows the modern tendency to enclose all gears in oil-tight cases or gear boxes and also provides a convenient method of introducing a third gear reduction. The previous designs did not permit as great a ratio of reduction between the motor and trackwheel as is frequently desirable.

Figure 4 shows the introduction of eight wheels instead of four, but retains the other features of Figure 1. It might be designated as the eight wheel, single motor drive of the "overhung type." Drives, as shown in Figures 2 or 3, could be applied as well. The reason for using the eight wheel design would be to secure lower wheel loads so as to keep within suitable limits based on axle bearing capacity, proper ratio between trackwheel diameter and rail width and in some cases the distribution of the loads to suit the strength of the runway.

Figure 5 shows an eight wheel bridge having four of the eight wheels driven; two of the wheels being driven

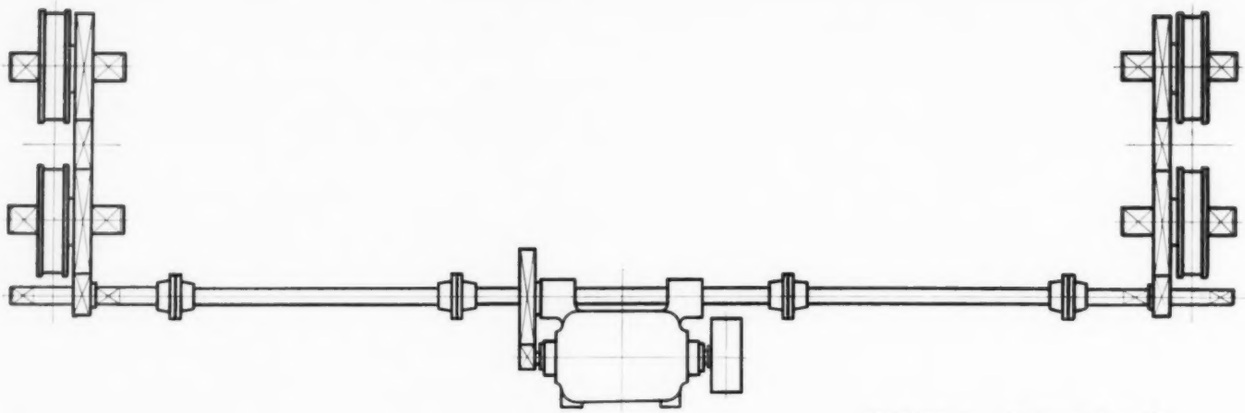


FIGURE 5. (Drive side only)

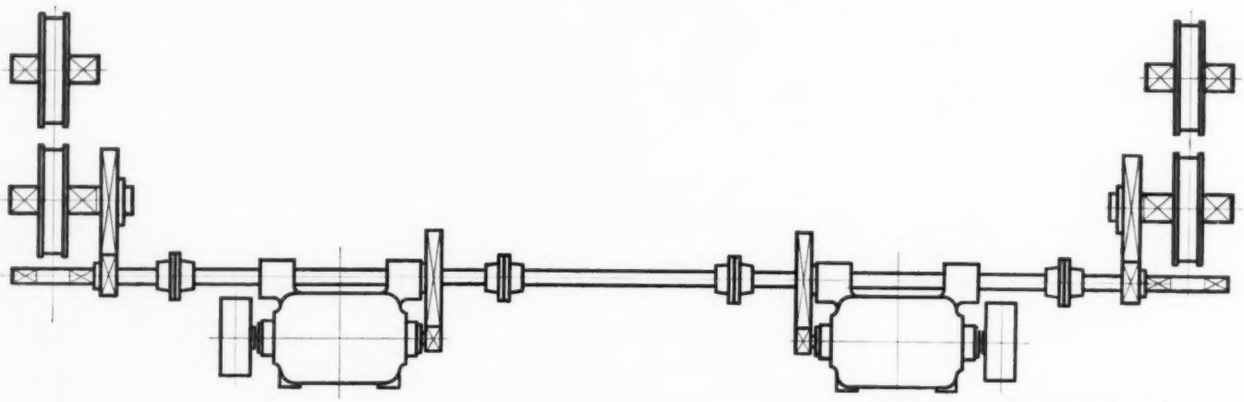


FIGURE 6. (Drive side only)

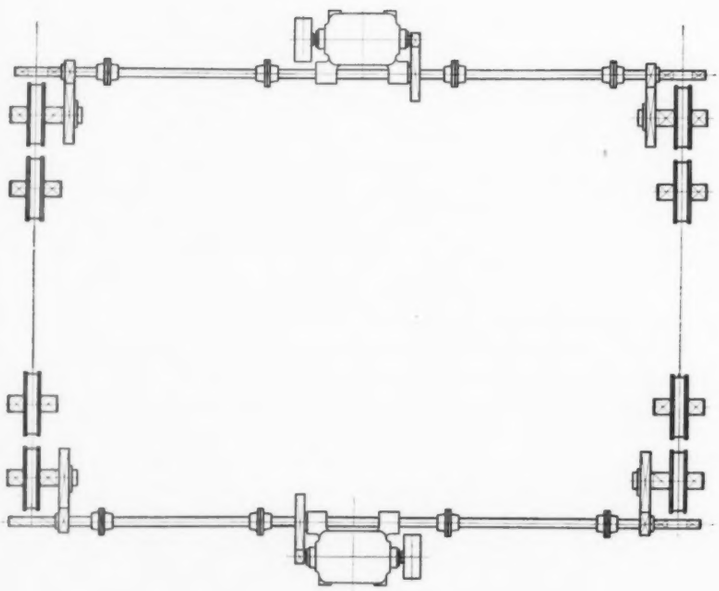


FIGURE 7

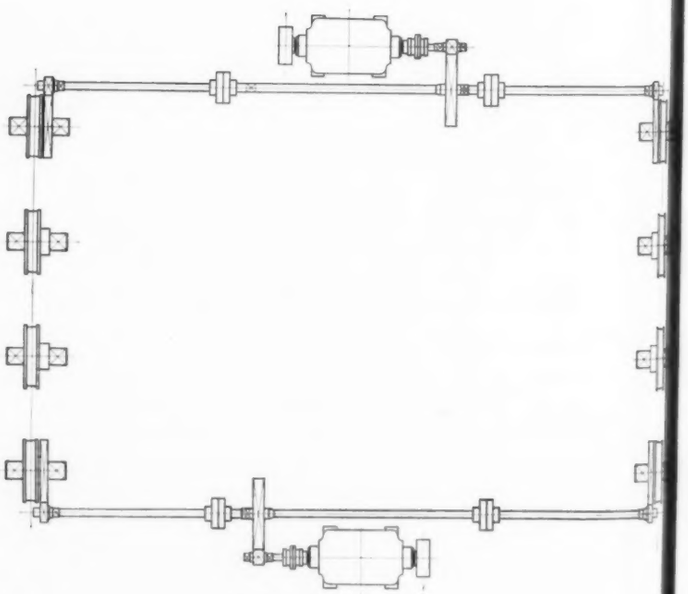


FIGURE 8

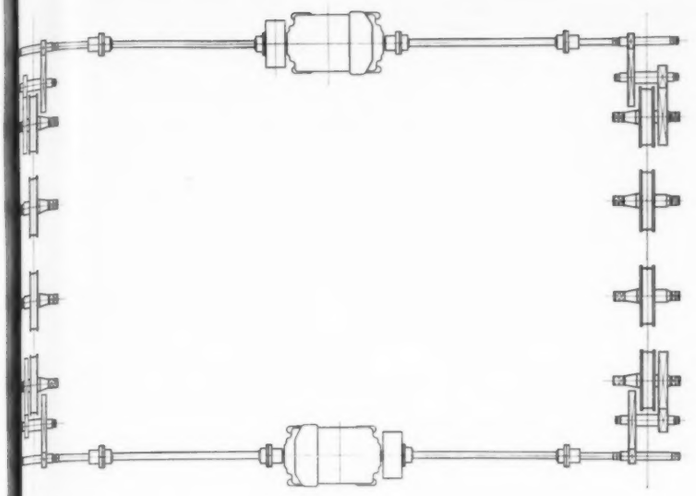


FIGURE 9

through idler gears. Two driven wheels are mounted in an equalizing truck at each end of the bridge. The track wheel driver gears are of the inside drive type and a single motor with one gear reduction at the center of the bridge span is used as in Figure 2. The reasons for adopting this type of drive will be discussed later.

Figure 6 shows the introduction of the double motor drive. A motor is located near each end of the bridge. An eight wheel bridge is shown with two driven trackwheels of the over-hung type. Of course the double motor idea could be applied as well to the inside drive type. This double motor drive may be chosen in order to provide continuous service in case of failure of one of the motors. Either motor should be capable of driving the bridge as an emergency operation while the other motor is being repaired. The arrangement may also be used when it is desired to connect the two motors in series for operation at half of their normal speed.

Figure 7 shows the more usual method of applying two motors to the bridge drive of a crane. This is known as the double bridge drive. It is usually applied to eight wheel cranes and two wheels are driven on each side of the crane. Either the overhung type or the inside drive type may be used with this arrangement.

Figure 8 shows one of the more modern arrangements of the double drive applied to an eight wheel crane of the inside drive type and with two wheels driven on each side of the crane. This design differs from Figure 7 in having the inside drive arrangement and in having the motor pinion mounted in a gear box and connected to the motor by a flexible coupling. It avoids the conventional arrangement of the motor gears on a horizontal center line and permits placing the bridge motor directly above the bridge line shaft and so supporting its weight much closer to the center line of the bridge girder.

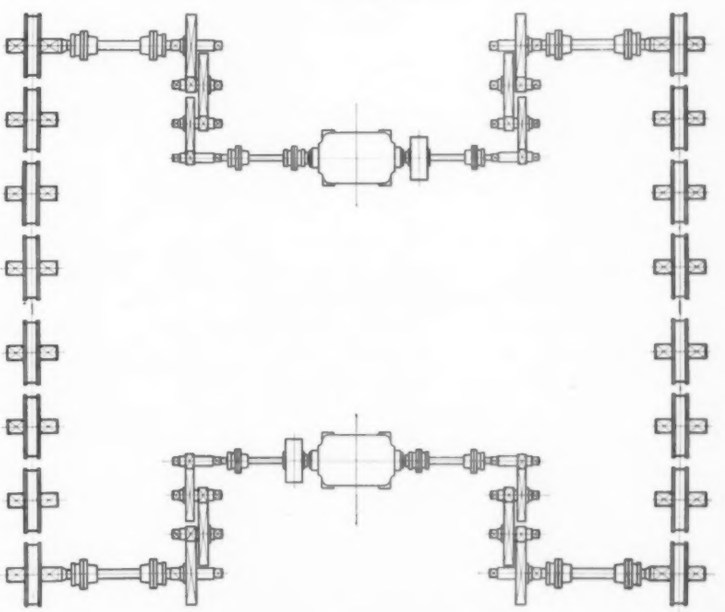
Figure 9 shows another modern arrangement of the double bridge drive with four trackwheels out of eight being driven. The motors are directly connected to the line shafts with flexible couplings. The gears are all mounted in the trucks and the high speed shafts of these gear reductions are connected to the line shaft through flexible couplings. This arrangement has the same advantage in regard to the weight of the motor being carried near the center line of the girder and also has some advantage in the fact that much smaller torque is carried by the line shaft which can therefore be lighter. This type of drive was developed by our company, I believe, about the year 1909 on the six girder type, hot metal cranes.

Figure 10 shows a drive with sixteen wheels and only two wheels driven on each side of the crane or four out of sixteen wheels. There are two wheels in each truck, which are of the equalizing type and there are two trucks connected to one equalizer at each corner of the crane.

All gears are enclosed in four gear boxes or speed reducers. The motors are mounted at the center of the bridge and directly connected to the line shaft which in turn is connected to the high speed shaft of the reducers by means of flexible couplings. The low speed shafts of the reducers are connected by flexible couplings to the trackwheel axles. This drive includes the most modern principles of construction with roller bearings throughout and hardened or heat treated gears enclosed in a dust-proof and oil-tight box. Suitable closures are provided at all shaft openings of gear boxes or bearing housings so as to retain the lubricant and prevent the entrance of dust.

Figure 11 shows another type of drive for a sixteen wheel crane and shows two motors driving four out of the sixteen wheels in a quite different arrangement. Each of the two driven trucks is provided with a complete individual drive which is mounted entirely upon the truck. There is no line shaft or squaring shaft connecting the two ends of the crane. This drive has proved especially satisfactory on certain types of gantry

FIGURE 10



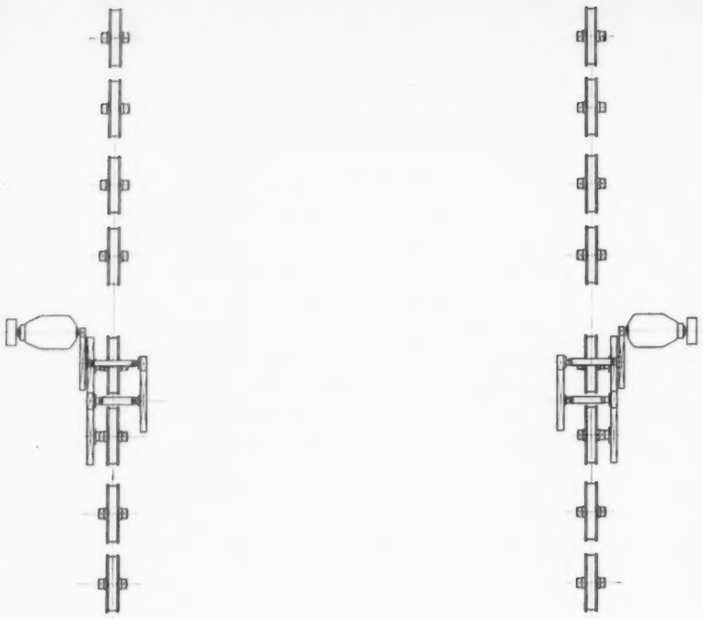


FIGURE 11

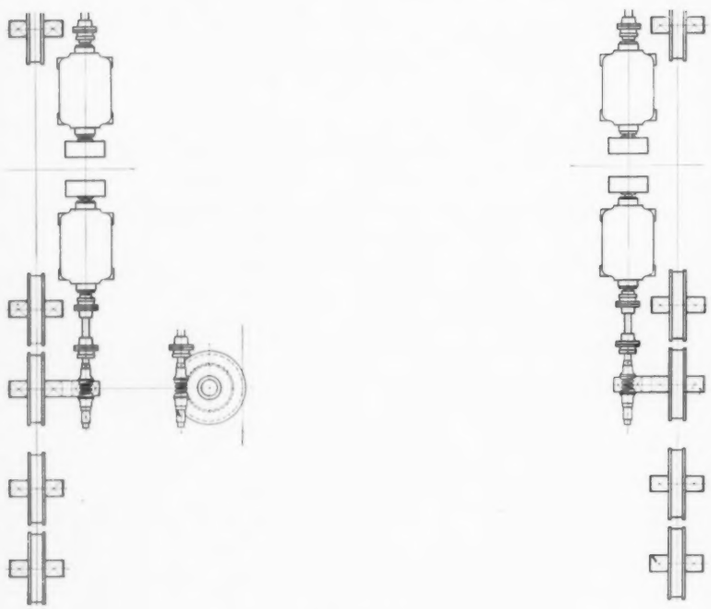


FIGURE 12

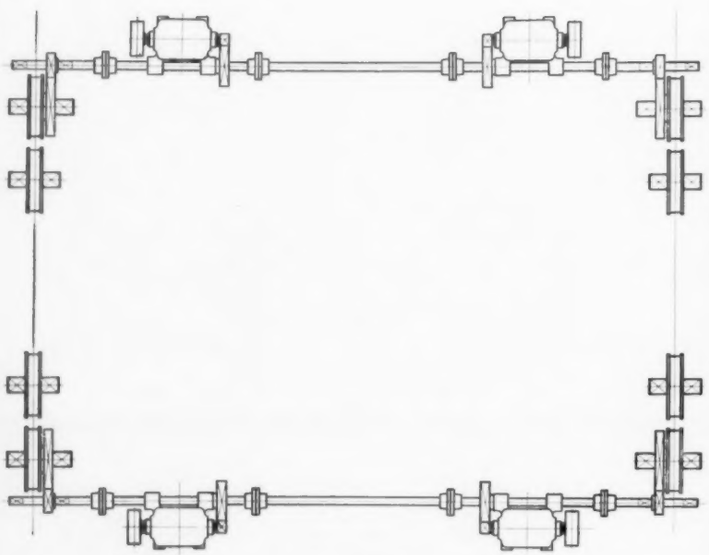


FIGURE 13

cranes where the elimination of the squaring shaft is a distinct advantage. We call this our direct motor drive.

Figure 12 shows a similar sixteen wheel bridge with the same number of driven wheels, that is, four out of sixteen. However, in this case four motors are used and one wheel on each of four trucks is driven by means of a worm gear reducer. We call this our "worm gear direct motor drive." Each driven truck supports the complete drive consisting of motor, brake and worm gear reducer. The motors are direct connected to the worm shafts by flexible couplings. With this arrangement one or even two of the motors may be removed temporarily for repairs without interruption of service.

The application of four motors to a bridge drive is also shown in the more conventional manner in Figure 13. This drawing shows the application of two double motor single drives to an eight wheel double drive crane.

In designing bridge drives with squaring shafts, these shafts should be designed to operate under extreme conditions of carrying the loaded trolley near one end of the bridge without exceeding a torsional deflection of about .08 degrees per linear foot.

We have developed a curve sheet (Figure 14) for reading the diameters of shafts according to the horsepower required. The line shaft couplings should be located near the bearing pedestals. Couplings are usually of the flange type and are covered by steel guards.

From past records, we find that the double bridge drive is usually required for cranes ninety feet span and above, although in some cases cranes having spans of seventy feet and above have two motors driving one squaring shaft, one motor on each end of the shaft located about fifteen to twenty feet from the center of the trackwheels.

The standard mill type crane for capacities from five tons to thirty-five tons usually have four wheels; cranes having a capacity of forty tons and up to one-hundred tons are usually eight wheel cranes; one hundred and fifty tons and above usually have sixteen wheels.

One of the problems which arises in connection with the number of driven wheels is how to overcome slipping of the drivers or lack of traction, when quick acceleration is wanted. The traction or coefficient of friction between the rail and a driven wheel may be taken to be 20 per cent of the load on that wheel. This represents a definite amount of power which can be applied to putting the crane in motion. If more power is supplied by the motor than this limiting amount, the trackwheel will spin around on the rail, but the crane will not move any faster.

Now this power which is limited by the amount of traction must be sufficient to provide acceleration, to overcome the rolling friction and also to overcome the axle bearing friction of all the wheels that are not driven. In this connection the item of axle bearing friction becomes important.

As long as the total weight of the loaded crane does not exceed certain limits the crane will ordinarily be built with just four bridge wheels and half of the wheels will be driven wheels. In this case the axle bearing friction of the two idler wheels is small in comparison with the traction on the two driven wheels, and there is no problem at all in regard to slipping.

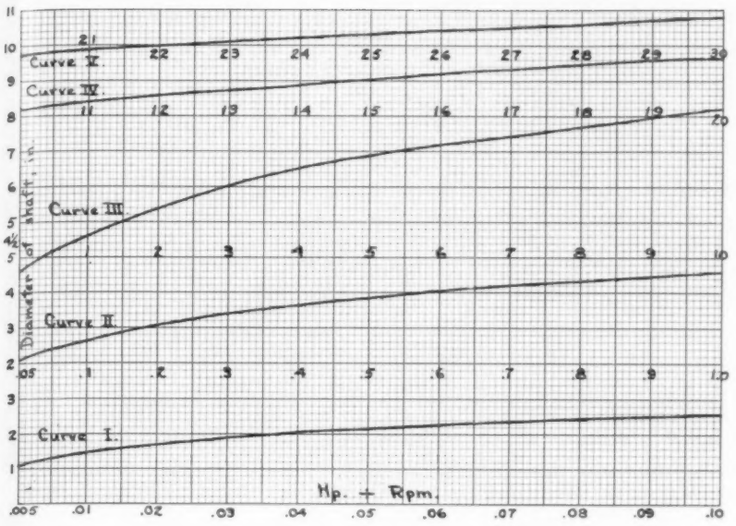


FIGURE 14

However, when either the span or the capacity is increased it becomes necessary to decide whether the wheel loads are too great if only four wheels are used. There are a number of factors entering into this decision, such as the capacity of axle bearings, the available end room in the building, the size of bridge rails which may be already installed. Also the matter of duplication of parts may be considered.

Having decided upon a bridge with eight wheels, it then becomes necessary to decide whether two or four of them should be driven. Here again the problem frequently solves itself because it is desired to have a drive on each side of the crane so that either drive alone can be used to operate the crane in case of breakdown of some part of one drive. However, a great many eight wheel cranes have been built having single bridge drives and we have found some rather interesting data concerning them.

Of 108 cranes of this type, one of which was equipped with anti-friction bearings, twenty-seven were built with four out of eight wheels driven and eighty-one had only two out of eight wheels driven. No case was found where slipping or lack of traction was very troublesome as long as the ratio of wheel diameter to axle diameter was at least 4.

Many cranes were equipped with sanders in order to keep the traction condition up to normal at all times or where quick acceleration made the margin small.

When still larger cranes were designed, it became necessary in many cases to use sixteen wheels. Here our experience with eight wheel cranes showed us that in most cases it is sufficient if one-fourth of the total number of wheels are driven. We found it important in this connection to reduce the axle bearing friction at the idler wheels. This could be done in the case of plain bearings by keeping the ratio of trackwheel diameter to axle diameter large. With the introduction of roller bearings the problem again tended to disappear.

With sixteen wheel cranes we believe that there are few if any cases where it is desirable to drive more than four of the sixteen wheels. This is readily accomplished with a drive on each side of the crane driving one wheel of each outside truck. In some cases where the second wheel has been driven through an idler pinion, these extra pinions and gears have been troublesome and expensive without apparently providing any worthwhile advantage.

The introduction of roller bearings on trackwheel axles has helped to eliminate wheel slippage during starting and acceleration. We must not overlook the fact that the tendency to slide driver wheels when braking has been increased. Braking action is more powerful due to the improved hydraulic brakes being used and crane bridges must be controlled at higher speeds than formerly, all of which has tended to transfer the problem of driver slippage from one of starting and acceleration to that of braking and stopping.

PERMISSIBLE RATES OF ACCELERATION

By

W. C. HEINLE

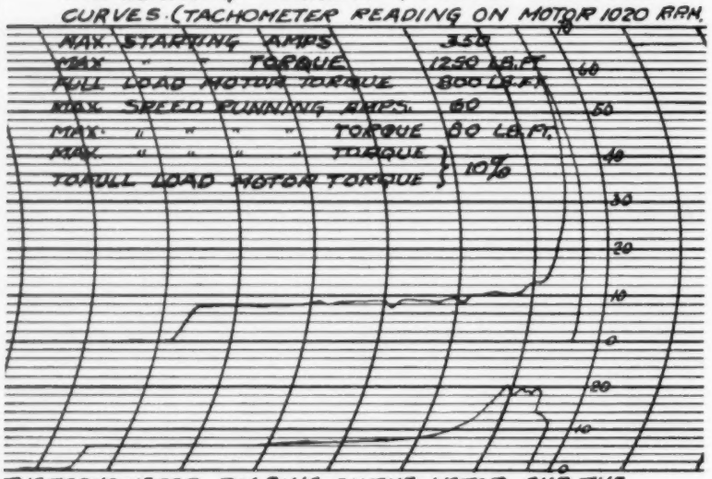
▲ I HAVE been asked to speak on the various factors that determine the proper rate of acceleration of crane bridge travel. To-day, I think it is generally true that steel mill operators specify approximately the same size bridge motor for a roller bearing crane that was formerly used on a bronze bearing crane. We have felt for a good many years that our bronze bearing experience would have to be discarded and an entirely new set of values and factors set up for roller bearing cranes.

The factors which must be considered in determining the proper rate of acceleration of crane bridge travel are: swinging of the load, track wheel slippage, and length of average crane movements. Proper rate of acceleration is difficult to analyze, due to the many combinations of weights and length of cables from the drum to hook. Obviously, the faster the initial rate of acceleration, the greater will be the hook swing. Once the size of the drive motor and gear ratio are fixed, the maximum rate of acceleration is determined by the resistor layout, the number of accelerating contacts and the setting of the timing relays. If this setting is such that it can cause excessive demands on the motor, load swing, or wheel slippage, the proper acceleration is then left entirely up to the judgment and skill of the operator.

While it may be desirable to have a rapid rate of acceleration without load, leaving it up to the operator, when handling a load, to keep it from swinging, there does not seem to be any logical reason for having the setting such as to cause excessive demands on the motor, particularly where the average movements are short and rather continuous.

Very high rates of acceleration, while on the resistor, may be a disadvantage and actually lose time on long runs. This will probably be more clearly shown by a

30 TON 120'-0 SPAN CRANE.
350 F.P.M. BRIDGE TRAVEL SPECIFIED.
620 F.P.M. " " " " ATTAINED.
TWO M.D.412 65 HP @ 430 R.P.M. MOTORS.
THE RUNNING CURRENT & TORQUE TAKEN FROM CURVES. (TACHOMETER READING ON MOTOR 1020 R.P.M.


MAX. STARTING AMPS 350
MAX. " " TORQUE 1250 LB.FT.
FULL LOAD MOTOR TORQUE 800 LB.FT.
MAX. SPEED RUNNING AMPS. 60
MAX. " " " " TORQUE 80 LB.FT.
MAX. " " " " TORQUE } 10%
TO FULL LOAD MOTOR TORQUE }

THE TACHOMETER READING ON THE MOTOR AND THE CURRENT READING DO NOT CHECK. THIS SEEMS TO INDICATE THAT ONE MOTOR IS DOING MORE OF THE WORK. THE CURRENT READING TAKEN ON ONE MOTOR

FIGURE 1

10 TON 80'-0" SPAN CRANE (MACHINE SHOP)
350 F.P.M. BRIDGE TRAVEL SPECIFIED.
480 F.P.M. " " " " ATTAINED.
ONE M.D.410 45 HP @ 460 R.P.M.

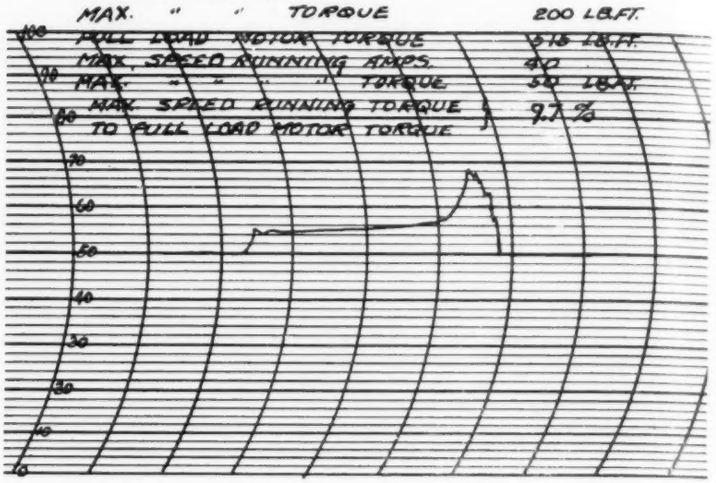

MAX. STARTING AMPS. 90
MAX. " " TORQUE 200 LB.FT.
FULL LOAD MOTOR TORQUE 215 LB.FT.
MAX. SPEED RUNNING AMPS. 40
MAX. " " " TORQUE 50 LB.FT.
MAX SPEED RUNNING TORQUE } 9.7 %
TO FULL LOAD MOTOR TORQUE }

FIGURE 3

15 TON 120'-0" SPAN CRANE.
500 F.P.M. BRIDGE TRAVEL SPECIFIED.
640 F.P.M. " " " " ATTAINED.
TWO M.D.410 · 45 HP @ 460 R.P.M.
READINGS TAKEN ON ONE MOTOR

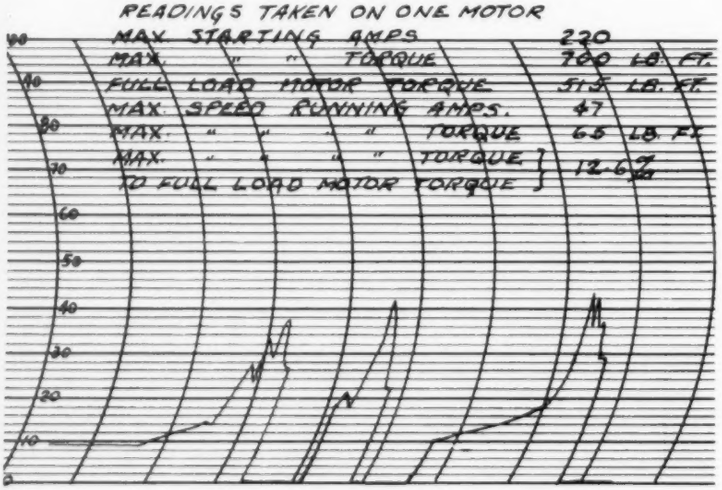
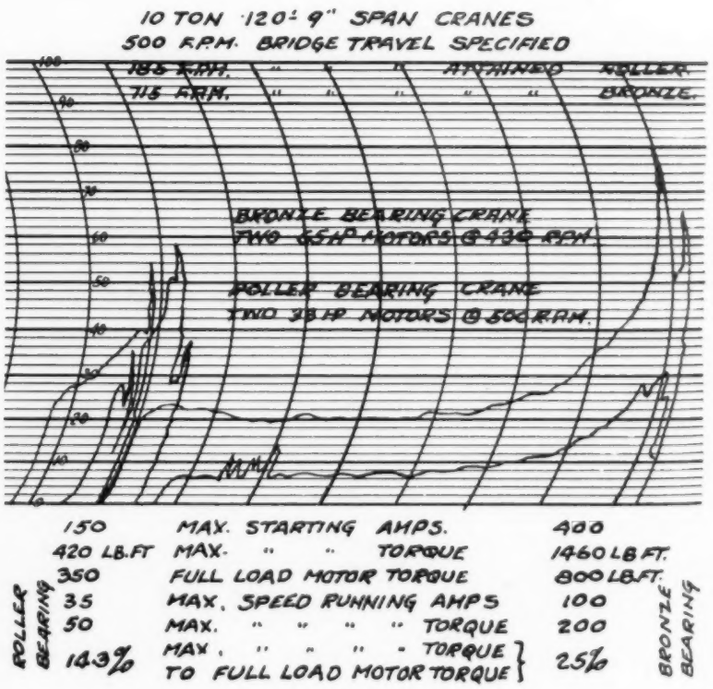

MAX. STARTING AMPS 220
MAX. " " TORQUE 760 LB. FT.
FULL LOAD MOTOR TORQUE 515 LB. FT.
MAX. SPEED RUNNING AMPS. 47
MAX. " " " " TORQUE 65 LB. FT.
MAX. " " " " TORQUE } 12.6 %
TO FULL LOAD MOTOR TORQUE }

FIGURE 2

10 TON 120'-9" SPAN CRANES
500 F.P.M. BRIDGE TRAVEL SPECIFIED


185 F.P.M. " " ATTAINED ROLLER
715 F.P.M. " " " " " BRONZE.
BRONZE BEARING CRANE
TWO 65 HP MOTORS @ 430 RPM.
ROLLER BEARING CRANE
TWO 38 HP MOTORS @ 500 R.P.M.

ROLLER BEARING		BRONZE BEARING
150	MAX. STARTING AMPS.	400
420 LB.FT	MAX. " " TORQUE	1460 LB.FT.
350	FULL LOAD MOTOR TORQUE	800 LB.FT.
35	MAX. SPEED RUNNING AMPS	100
50	MAX. " " " " TORQUE	200
14.3%	MAX. " " " " TORQUE TO FULL LOAD MOTOR TORQUE	25%

FIGURE 4

later speaker. Slippage is not likely to occur on a crane with one-half the wheels driven unless the rails are wet or greasy. It is possible on eight wheel cranes, where very rapid acceleration is not necessary, to drive only two wheels, although the final running speed may be high.

Figure 1 was taken on a 30-ton 120-ft. span crane driven by two MD-412, 65 hp. at 430 rpm. motors. The specified bridge speed was 350 ft. per min. Due to the large motors used, it was not practical to gear down to this speed. Consequently the maximum running speed proved to be 620 ft. per min. taken from the speed of the motor which was 1000 rpm.

We have shown two bridge movements, one above the other, which clearly illustrate what happens. In the top figure the operator quickly moved the master to the maximum position, and the setting was such that the contacts all closed in less than one second, which caused excessive load swing, as can be noted by the wavy lines on the chart. The lower figure indicates the maximum acceleration possible to keep the load from swinging.

This is a case where the setting should have been such as to prevent this excessive acceleration.

The current reading was taken on one motor only. In one direction the chart indicated 40 amperes, in the other direction 35 amperes.

This checked entirely too low when considering the speed of the motor, which seemed to indicate that this motor was not doing its proper share of the work, but in further checking the chart it was found that the indicator had dropped considerably below the zero line when idling. If it had been properly set, the chart would have shown between 55 and 60 amperes, which checks closely with the speed of the motor.

The full load torque of this motor is 800 ft.-lb. The maximum starting current of 350 amperes is equal to approximately 1250 ft.-lb., which is slightly more than 50 per cent above full load torque. The current at full speed is 60 amperes, which represents 80 ft.-lb., or approximately 10 per cent of full load torque.

Since 350 ft. per min. was specified for this crane, it is obvious that two 33 hp. motors would have been more satisfactory, both from the standpoint of control and motor efficiency.

Figure 2 shows a 15-ton 120-ft. span crane driven by two MD-410, 45 hp. motors at 460 rpm. The specified speed was 500 ft. per min., but the actual running speed was 637 ft. per min. The tachometer speed of the motor and the current readings on the chart checked very closely.

This crane is on a long runway, with a number of other cranes. The average travel was comparatively short and frequent. In fact, it was necessary to wait quite some time before there was enough space available to get it up to the maximum speed. The full load torque of this motor is 515 ft.-lb. The worst condition of acceleration shown indicates a peak torque of 700 ft.-lb., which is only 36 per cent above the full load torque. The maximum running torque is 65 ft.-lb., which is only 12.6 per cent of the full load torque. It is possible that 400 ft. per min. maximum running speed with two 33 hp. motors would work satisfactorily on this crane.

Figure 3 shows a 10-ton, 80 ft. span machine shop crane, driven by an MD-410, 45 hp. at 460 rpm. motor. The specified bridge speed 350 ft. per min., the maximum running speed obtained 460 ft. per min. The maximum running speed of the motor was 1200 rpm., and the chart indicates 35 amperes, or approximately 35 ft.-lb. torque. As the switchboard showed 248 volts, this more nearly represents a torque of 50 ft.-lb. To be conservative, we will use this figure.

The full load torque of the motor is 515 ft.-lb. The maximum starting torque is 200 ft.-lb.; the maximum running torque is 50 ft.-lb., which is only 9.7 per cent of the full load torque of the motor.

A 19 hp. motor would meet all the requirements of this crane, as it has intermittent duty and rate of acceleration is not a factor.

About fourteen years ago a steel mill in the Pittsburgh district purchased a number of cranes. As they had not used any roller bearing cranes up to that time, it was decided to purchase several, and place them on the same runway with cranes equipped with bronze bearings.

Figure 4 shows the relation between two 10-ton, 120 ft. span cranes on the same runway, the cranes being identical, except the bearings. The roller bearing crane was equipped with two 33 hp. motors, the bronze bearing crane with two 65 hp. motors. The specified bridge speed was 500 ft. per min. The actual speed of the roller bearing crane proved to be 785 ft. per min., the bronze bearing 715 ft. per min.

At that time, we knew that the power requirements for a roller bearing crane were less than one-half of that required for a bronze bearing crane, which the chart proves. The current readings of the graph checked out closely to the tachometer speeds of the motors.

The full load torque of the 33 hp. motors is 350 ft.-lb. The maximum starting torque is 420 ft.-lb.; the maximum running torque is 50 ft.-lb., which is only 14.3 per cent of the full load torque.

The maximum starting torque on the bronze bearing crane was 1460 ft.-lb. This could, however, be adjusted to a lower figure, and not interfere with the operation of the crane. The full load torque of the 65 hp. motor is 800 ft.-lb., the maximum running torque 200, which is only 25 per cent of the full load torque. Both of these cranes were stopped by plugging the motors. However, with the perfected type of foot brake available to-day, this should not be necessary unless some extreme emergency arises.

These charts definitely show: that the power required to drive a roller bearing crane is much less than that to drive a bronze bearing crane; that the percentage of static torque to rolling torque is less on roller bearings than on bronze bearings; and that with the proper foot brake to stop the crane, no demand is made on the motor for this purpose.

Since it is true that the application of over-size motors on roller bearing cranes produces excessive acceleration and high ultimate bridge motor speed, operating at a low efficiency, it is evident that more consideration should be given to the use of smaller motors.

COEFFICIENTS OF ROLLING FRICTION

By

C. BRONGERSMA

▲ FOR many years we have made use of a formula for rolling friction which takes into account the diameter of the bridge wheel axle, the diameter of the bridge wheel, the coefficient of friction, the efficiency of the gearing. These factors are combined into one unit which we have labeled "traction." The formula for traction is: Traction equals weight of crane plus load in pounds, times the diameter of the axle divided by the diameter of the wheel, times the coefficient of friction, divided by the efficiency of the gearing. For bronze bushed cranes we use a figure for coefficient of friction equal to .08. For roller bearings we use .035. Traction =

$$\frac{(\text{Wt. of crane} + \text{load in lb.}) \times \text{axle dia.} \times \text{coeff. of friction}}{\text{Wheel diameter} \times \text{efficiency of gearing}}$$

Gear efficiency:

2 reductions	80 per cent
3 reductions	70 per cent
2 reductions and idler	75 per cent

The horsepower consumed equals traction times the speed of crane in ft. per min. divided by 33,000.

$$\text{Hp. consumed} = \frac{\text{Ft. per min. speed} \times \text{traction}}{33000}$$

For direct current cranes, select motor with at least 25 per cent excess horsepower above the figures from above formula. Then go to motor characteristic curve and pick out motor speed in rpm. from curve for the above consumed horsepower. The gear ratio required then equals the rpm. of the motor times the circumference of the wheel in feet, divided by the speed of the crane in feet per minute. At the same time, the motor torque equals the traction times the radius of the wheel in feet divided by the gear ratio.

Gear ratio = Motor rpm. from curve at consumed hp. × Circumference of wheel in ft. ÷ Ft. per min. crane speed.

$$\text{Motor torque} = \frac{\text{Traction} \times \text{radius of wheel in feet}}{\text{Gear ratio}}$$

Figured in this manner, the free running speed of the crane will be near to the specified speed. The excess horsepower will be used for quicker acceleration, and the crane will be quicker on short movements, than a similar crane where the gearing is figured to suit full load speed of the motor instead of actual motor speed from motor characteristic curves.

In the above formula the diameter of axle is taken as the mean diameter of the roller circle for roller bearing cranes, and as the true diameter of the axle for a bronze bushed job. The figures of .08 for bronze bearings and .035 for roller bearings hold true for bridge wheels with straight treads running on good runways. We have run tests where we compared a roller bearing crane and a bronze bushed crane on the same runway. These tests indicate that the above figures are conservative and that the cranes will always make their specified speed using these values. The figures would undoubtedly be modified downward for taper tread wheels, as taper tread wheels remove some of the variable flange friction from factors entering into the friction coefficient.

The term "traction" in the above formula is really equal to the pounds per ton rolling friction, times the weight of the crane plus the load in tons, divided by the gearing efficiency. The formula simply recognizes the different results experienced when the ratio of wheel diameter to axle diameter is changed. Some tests made recently with taper tread wheels proved that the figures for coefficient of friction can safely be reduced. Some figures for roller bearing, taper tread wheels show a value of 10 to 14 lb. per ton rolling friction not calculating gear efficiencies. These figures will give a friction coefficient approximately one-half the above values.

The use of nitrided or hardened pins and burnished bushings naturally results in a lower coefficient of friction than .08 but at present we have no test results to check this.

In actual practice there is a tendency to over-motor bridges on steel mill cranes. The motors are usually so large compared to the friction load that tests are not very accurate. This over-motoring is probably due to the frequent plugging of motors, to variations in runway conditions, frequency of operation, high ambient temperatures, etc., not considered in the above formula.

Because of these variations, we have adhered to the higher coefficient of friction mentioned above, even though we realize that roller bearing taper tread wheels, etc., actually give a lower figure. As you all know a large percentage of the steel concerns specify the size of motors to be used and we as crane builders naturally assume that the reason for the size chosen is actual operating experience, which has proven a certain size to be necessary.

INFLUENCE OF BEARING TYPES ON JOURNAL FRICTION

By

E. C. RICE

▲ WHILE we generally think that journal friction of a crane depends primarily on the type of bearing used, we should also consider the influence obtained by design, manufacture, lubrication and maintenance of the type selected.

Fundamentally, sleeve, roller and pin bearings are classified into two groups, depending on the method of eliminating or reducing sliding friction. Sleeve and pin bearings belong to the group where sliding friction is reduced or eliminated by separating the bearing surfaces with a fluid. Roller bearings are included in the group where friction is reduced by separating the sliding surfaces with rollers. In other words, transferring sliding motion to rolling motion.

In crane operation we have three stages, starting, running and stopping. Since the majority of cranes

have relatively short travels and are started and stopped many times a day, the starting period is very important in considering the influence of the type of bearing used.

Sleeve and pin bearings depend on the maintenance of an oil film to obtain low friction values. However, when the crane is stopped, the oil film is broken and the starting or break away resistance is high when compared with a roller bearing, which has practically uniform resistance for all speeds or conditions.

Numerous tests by the manufacturers of roller bearings on railway cars indicate that the break away resistance of a roller bearing is approximately one-tenth that of a sleeve bearing. We have demonstrated many times the low break away resistance of a roller bearing crane by having a man push a crane weighing approximately twenty-seven tons back and forth on our runway.

The length of time required to build up a satisfactory oil film on a crane equipped with sleeve or pin bearings is difficult to determine. It would seem, however, that the time required is not of sufficient length to influence greatly the overall power requirements of the crane. However, the higher break away resistance will definitely affect the starting torque required, and the size of motor and gear ratio selected.

In the next stage of the cycle of operation, namely that of running, the influence of the type of bearing is not so great. The tests on railway cars indicate that the draw bar pull for roller bearings is about 10 per cent less than the draw bar pull of sleeve bearings.

In a test made January 7, 1919, at the plant of the Erie Steel Construction Company, it was found that the running draw bar pull for the roller bearing crane was about 50 per cent of that required for a plain bearing crane.

In 1929 we made a comparative test on two cranes, one equipped with pin bearings and the other equipped with roller bearings. This test indicated that the roller bearing crane required about half the tractive effort required by the sleeve bearing crane. We do not contend that these two tests prove conclusively that this is the ratio between the two types of bearings but we do feel they indicate a distinct advantage in favor of a roller bearing.

During the last cycle of operation, that of stopping a crane, the advantages are reversed as far as braking effort required. A roller bearing crane will require a greater braking torque to stop in a specified distance than is required by the sleeve bearing or pin bearing crane. This torque may be obtained by the use of a brake or by plugging the motor. With manual control, plugging is not used extensively by the average operation. On the other hand, plugging is used extensively with magnetic control and is a factor to be considered in determining the size of motor required.

A sleeve bearing will give some kind of service irrespective of the attention it receives. In other words, it can be greatly neglected in lubrication, maintenance and protection against dirt, and still give a great deal of service although the power consumption will be high. This may account for the poor maintenance that sleeve bearings sometimes receive. I have seen a set of gears on a job crane that jumped about ½ in. every time the motor was reversed but the crane kept on running.

If these adverse conditions cannot be corrected, then a sleeve bearing should be used, as a roller bearing would soon fail completely. Design, workmanship and maintenance on cranes today has eliminated these unfavorable conditions so that roller bearings give very satisfactory service and reduced power costs.

A roller bearing must not be too tight or trouble will quickly develop. Tightness causes the rollers to spiral, resulting in end thrust and power loss as well as rapid wear on the bearing. Tightness also causes heating which in turn increases the tightness and accelerates the wear and power loss. Bearings should have sufficient clearance so that when the rollers leave the load zone they are free to adjust themselves as to alignment and spacing.

Before the advent of roller bearings in crane design, it was common practice to use 60 lb. per ton as the draw bar pull for figuring the size of bridge motors. Another formula took into account the ratio of bridge axle diameter to wheel diameter. If this ratio was somewhere between 4 and 5 the draw bar factor became approximately 60 lb. per ton.

On the first roller bearing cranes we used approximately the same formula. After a few installations had been made we ran the test mentioned earlier in the discussion and developed the following formula:

$$\text{Hp.} = \frac{\text{Weight tons} \times V \times \left(\frac{30 + V}{20}\right)}{33000}$$

With a bridge speed of 300 ft. per min. this formula gave a draw bar pull of 45 lb. per ton.

We recently made a test on a 10-ton crane with load which was designed to run 413 ft. per min. We found that this crane would accelerate at the rate of 2.75 ft. per sec. per sec. which indicated that the friction load was very low. A deceleration test on same crane resulted in a rate of .15 ft. per sec. per sec. which is equivalent to a draw bar pull of approximately 5 lb. per ton.

However, this must be modified by the fly wheel effect of the motor and brake wheel and couplings so that the actual deceleration due to journal friction would be somewhat higher than the curve indicated.

When roller bearings are properly applied there is a distinct savings in motor requirements, due to the much lower starting friction than with sleeve or pin bearings.

The motor size is more dependent upon starting conditions than it is upon running conditions. The power required to accelerate a crane is appreciable but the power required to keep the crane in motion is relatively small. Since the power required to accelerate is high it is very desirable to keep the starting friction as low as possible. This is where the roller bearing has a distinct advantage over the sleeve or pin bearing.

Since the majority of the cranes manufactured today are equipped with roller bearings, we feel that we need not give a great deal of consideration to the comparative influence of types of bearings but we do think we can learn more about journal friction of the roller bearings which will aid us in selecting the motor, brake and control equipment best suited for each installation. We also feel that a more comprehensive study should be made regarding the normal requirements of each crane installation, if the overall cost of crane service is to be reduced.

REQUIREMENTS OF BUMPERS AND RAIL STOPS

By

R. J. WADD

▲ IN an analysis of the requirements for protection of crane bridges for movement between cranes or into runway rail stops we must take into account: first, the possible forces to be considered; second, the degree of protection necessary or desired, and third, the mechanism to be used to obtain protection.

For maximum bumper forces on hook type cranes we have the possibility of the kinetic energy of the crane bridge and trolley operating at maximum speed plus any effective force of the live load. The latter considered as a swinging load can be treated as a pendulum with due allowance for efficiency of the hoist tackle. The maximum effect of the live load can then only be the horizontal component as the kinetic energy of the load is absorbed by the gravity pull at the end of its swing. The load, starting to swing as the bridge slows down, reaches its amplitude some time after the bridge motion ceases and the energy of the bridge has been absorbed. The only live load effect will be the co-acting value at the time bridge motion ceases. For all practical conditions, it is impossible for this co-acting value to exceed or equal the value of the kinetic energy of the bridge, absorbed in a reasonable distance; therefore we need only to consider the bridge and trolley forces in our calculations.

In considering the amount of protection desired or necessary we must appreciate that the maximum conditions of loading, namely, collision under full load and speed between crane and runway rail stops are rather of remote possibility and involve high inertia values, especially on the heavy, high speed structures. Furthermore, the hazard of collision is not fixed but will vary with several factors such as number of cranes per runway and the local conditions of operation. For normal installations, the greatest possibility of collision exists between cranes; contact with track stops under maximum conditions of load and speed should be rather remote. In the modern mill or industrial plant equipped with any number of cranes, the operators are carefully selected as to fitness for the job and by their skill of operation, remove the maximum hazard of collision.

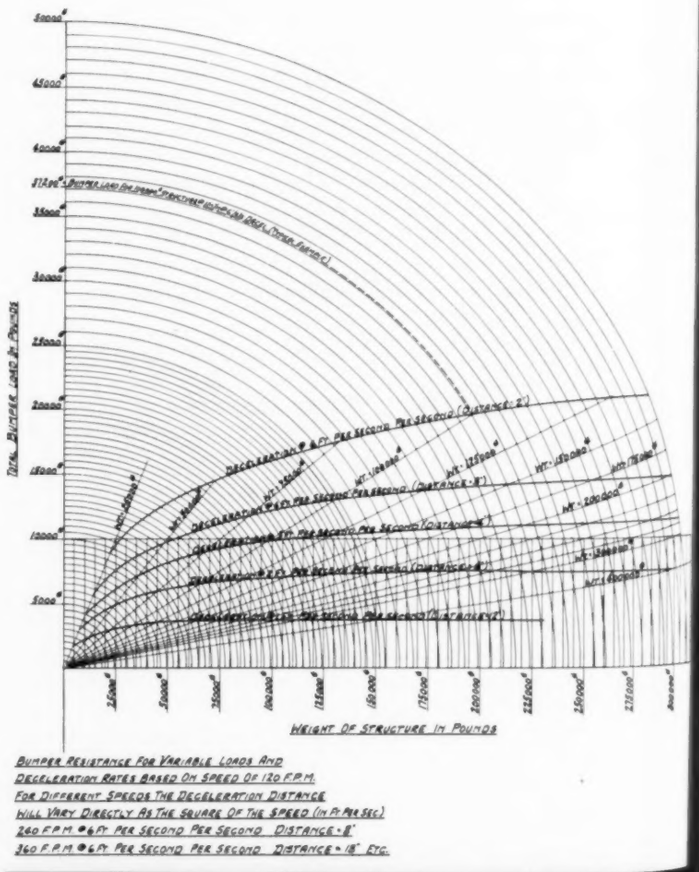
We therefore have the problem of providing maximum protection for only the extremely hazardous installations and the possibility of a compromise on the requirements for the average installation, or production cranes, dependent upon the operator to avoid dangerous conditions as in handling the modern high speed automobile. In the average installation the operator can easily keep his speed down to a slow speed control point when in the zone of possible danger.

In applying protection for full collision forces we must appreciate that the allowable space for deceleration may be pertinent to hook approach and will probably require a different rate of deceleration over a range of varied crane weights and speeds. The space factor for mounting the mechanism is also a considerarion. In establishing data for loading of the bumper device we must consider the structural strength of runway details and existing cranes. The maximum horizontal runway load under crane operation will be approximately 15 per cent to 25 per cent of the maximum wheel load of driven wheels of one or more cranes on the runway as the wheels are skidded in braking or in excessive acceleration. When possible, bumper mechanism should operate at a pressure within this limit in the design for contacting end stops. The operating pressure must also be within the limits of the structural strength of end truck details of cranes in collision or operating contact and also of a value that will not throw the trolley off the rails. Furthermore, the pressure must be within the limits of spring or draft gear details for space mounting and allowable deceleration distance.

It is not a very difficult problem to apply the necessary and proper protection for each individual installation, which is really the correct manner of procedure as it is rather impractical to attempt to apply a standard formula for the two conditions of hazard. The following graph charts W/G and $V^2/2$ at varying rates of deceleration and can be used to select practical spring or draft gear loads in various allowable deceleration distances.

In applying full protection to the slower moving

Graph for bumper spring selection for various loads and deceleration rates.


WEIGHT OF STRUCTURE IN POUNDS
BUMPER RESISTANCE FOR VARIABLE LOADS AND DECELERATION RATES BASED ON SPEED OF 120 F.P.M.
FOR DIFFERENT SPEEDS THE DECELERATION DISTANCE WILL VARY DIRECTLY AS THE SQUARE OF THE SPEED (IN FT PER SEC)
240 F.P.M. @ 6 FT. PER SECOND PER SECOND DISTANCE = 8"
360 F.P.M. @ 6 FT. PER SECOND PER SECOND DISTANCE = 18" ETC.

cranes up to about 300 ft. per min. a rate of deceleration of 6 ft. per sec. per sec. per 100,000 lb. approximate weight of structures is practical for space and loading. For increased weight of structure, the rate of deceleration should be proportionately decreased. For speeds exceeding 300 ft. per min. auxiliary slow down means are desirable such as limit switches actuating solenoid or dynamic braking mechanisms; the additional device preferably to actuate ahead of any mechanical mechanism and reducing the value of the kinetic energy to the capacity of the mechanical device.

In applying partial protection depending upon the operator to avoid the maximum hazard we can reduce the full load 50 per cent in determining the value for the kinetic energy and use a rate of deceleration as used in the collision condition. This will provide approximately 50 per cent of full collision protection as a minimum requirement and should be satisfactory as with careful operation the slow control speed can be obtained on a-c. as well as d-c. cranes.

The above would give the following limiting data. for full collision forces—37,000 lb. (18,500 lb. per crane end) will absorb kinetic energy of:

100,000 lb. weight of bridge and trolley at 6 ft. per sec. per sec. deceleration
- With crane speed of 120 ft. per min. deceleration distance = 2 in.
- With crane speed of 300 ft. per min. deceleration distance = 12½ in.

200,000 lb. weight of bridge and trolley at 3 ft. per sec. per sec. deceleration
- With crane speed of 120 ft. per min. deceleration distance = 4 in.
- With crane speed of 300 ft. per min. deceleration distance = 25 in.

300,000 lb. weight of bridge and trolley at 2 ft. per sec. per sec. deceleration
- With crane speed of 120 ft. per min. deceleration distance = 6 in.
- With crane speed of 300 ft. per min. deceleration distance = 37½ in.

For partial collision forces—18,500 lb. (9250 lb. per crane end) will absorb 50 per cent kinetic energy of—

100,000 lb. weight of crane and trolley at 6 ft. per sec. per sec. deceleration
- With crane speed of 120 ft. per min. deceleration distance = 2 in.
- With crane speed of 300 ft. per min. deceleration distance = 12½ in.
- With crane speed of 600 ft. per min. deceleration distance = 50 in.

200,000 lb. weight of crane and trolley at 3 ft. per sec. per sec. deceleration
- With crane speed of 120 ft. per min. deceleration distance = 4 in.
- With crane speed of 300 ft. per min. deceleration distance = 25 in.
- With crane speed of 600 ft. per min. deceleration distance = 100 in.

The foregoing is based on our own particular data as used for single action compression spring type bumper or combination rail stop for medium capacity units. It might be necessary to modify somewhat to adapt to other bumper devices such as various draft gear units. These figures show the impracticability of using the conventional single action compression spring unit for the heavy high speed cranes especially due to space factor.

The use of draft gear in place of compression spring actuated devices has certain advantages in their ability to build up pressure with the rate of deceleration and can thus offer practically 100 per cent collision protection without excessive ultimate pressure especially well adapted to the heavier capacity high speed units. Approximate equivalent results can be obtained by using compound acting compression spring units to produce the increasing pressure.

For the lighter weight cranes or conditions limited by runway or existing crane restrictions lower pressure values can and should be used. In applying protection to the slower moving light capacity bridges as well as for the trolley units it is practical to use combination stops with live block rubber insets contacting the wheel tread. They have demonstrated very good life and performance.

Our experience with limit switch apparatus for crane protection while not extensive, has not been entirely satisfactory. While we would consider it practical for auxiliary means, we do not believe it fool-proof enough to utilize as a standard method of 100 per cent protection.

Due to the many variable factors that enter into bumper calculations, the writer would suggest that crane specifications indicate the percentage of protection demanded and have the exact design data fit the practical limitations of application for that particular installation.

HORSEPOWER REQUIREMENTS AND MOTOR APPLICATION

By

G. A. CALDWELL

▲ THE greatest problem in selecting the proper size of motor for a crane bridge drive does not consist of making a lot of elaborate calculations, but in obtaining sufficient data on each particular crane and the duty it is to perform in order to make an intelligent application. Since the crane builder is usually more familiar with the mill layout and the requirements of the various cranes, they select the motor rating in cooperation with the ultimate customer. The electrical manufacturer is rarely consulted on these points, although quite often he asks for certain pertinent data such as weight, speed, type of bearing, etc., when designing the control, mainly as a check on the correct resistor values for the accelerating resistor and the number of accelerating points. However, the results of the calculations made for the design of the resistor and controller are in themselves an indication as to whether the bridge drive is overmotored or undermotored.

If an attempt is made to determine the methods used by the steel companies in determining the correct horse-

power for a given crane, you find the experience with previous cranes playing an important part. In a mill where there are a large number of cranes, it is usually possible to compare the proposed crane to a similar one already in operation. If it is about the same weight and speed and of similar mechanical construction, the selection of the proper motor for the new drive is simple. If the unit in service has a tendency to be sluggish and also has a motor that is overheating, it is obvious that the next larger rating is desirable. If on the other hand the crane has a very rapid rate of acceleration, excessive balancing speed, and a tendency for the wheels to slip on plugging, a smaller motor can be considered, especially if operating temperature of the motor is below the normal rating. But if the crane is satisfactory in regard to the points mentioned above, then there is certainly no reason for changing the size of the motor on the new installation.

I believe that you will find that most crane engineers have some formula that they use as a check on the conclusions reached on the basis of their previous experience. This formula always contains the factors of weight and speed multiplied by constants. However, the constants are also really a result of experience and will be changed to take care of such factors as the efficiency of gearing, type of bearings used on the axles, etc.

Our opinion is that satisfactory results can be obtained by this method so long as the speed of the bridge varies over a fairly small range or if the constant is automatically increased for increased bridge speed, and decreased for lower crane speed. This conclusion is verified by the observations that it is usually the high speed cranes running from 400 ft. per min. and higher that are burning out armatures and the larger heavy duty low speed hot metal cranes, etc., where the wheels are slipping on plugging.

If the problem of calculating the correct motor size for the bridge of a crane was given to an engineer who had no previous experience with the crane designs, he could make satisfactory calculations by using certain fundamental formulae with which he would be familiar. The data he would need are the weight of the crane, the distribution of weight on the wheels of the crane, the gear ratio, size of wheels, the desired speed of the bridge and the coefficient of friction in the axle bearings and the efficiencies of the various gear units.

If we assume that the crane is up to its normal running speed, then a certain amount of horsepower will be required to keep the crane running at a given speed, and this consists only in overcoming the friction of the drive. It is this value of horsepower that determines what the final balancing speed of the motor will be and if this value is only a small percentage of the motor rating, then the final balancing speed will be quite a bit higher than the calculated base speed on the rating of the motor at full load torque. This, of course, is due to the familiar characteristic of a series would motor which causes it to run at slow speeds with a heavy load and at high speeds under light load. This characteristic is both desirable and undesirable depending upon the viewpoint.

The horsepower required to overcome friction at a constant speed can be calculated from the formula as follows:

$$P = \frac{F \times W \times V}{33000}$$

where P = Power in horsepower.
F = Force to overcome friction in lb. per ton
W = Weight of crane in tons.
V = Speed in ft. per min.

The force to overcome friction expressed in lbs. per ton will of course vary over quite a wide range depending upon the type of bearing, the possible misalignment of wheels on the track, and the condition of the track. A few years ago, a figure of 30 lb. per ton was quite often used as a good average value for this figure. However, with the modern anti-friction bearing cranes of good design and with a good track on which to run, the figure is reduced to from 10 to 15 lb. per ton and the writer has even heard the figure of 8 lb. per ton used. The experience of one steel mill where they had a crane that was geared for approximately 225 ft. per min. with the motor operating at full load torque and where under test they found that it was actually running approximately 600 ft. per min. at its balancing speed indicates that the actual horsepower to drive the crane at constant speed is only a very small percentage of the rating of the motor.

The other portion of the load which is the maximum part consists of the work required to accelerate and decelerate the moving parts. This value can be calculated from the following formula:

$$\text{Work} = \frac{W \times S^2}{17{,}700}$$

W = Weight in pounds.
S = Speed in feet per second.

This formula will be recognized as the old friend $W = \frac{1}{2}MV^2$ and the important thing to note is that the work required to accelerate the crane varies not directly as the speed, but as the square of the speed. Therefore, if other things remain equal and the speed of the bridge drive is increased, the work required for acceleration increases as the square of the speed. It is my opinion that this relation has been overlooked in some cases, and explains the reason for overheated armatures of some of the high speed cranes that receive heavy duty service.

A typical crane for use in a wide strip mill will probably have a span from 80 to 100 ft., will be rated 15 tons and will probably have a total weight with load of 115 tons. If we assume that this crane is geared for a normal running speed of 250 ft. per min., the stored energy in the crane at that speed or the work that the motor must do to accelerate to that speed will be 226 hp. seconds. If we assume that the crane should be accelerated in 5 seconds then the torque which the motor must exert during the accelerating time is equivalent to 90 hp.

On this same crane, if we assume that the running friction is 15 lb. per ton, the horsepower required to keep it running at 250 ft. per min. will only be 13 hp. However, we know that the crane will require at least a 65 hp., ¼ hr. crane rated mill type motor with a full load speed of 440 rpm. to provide the accelerating torque

indicated above. However, at this light load, the motor will finally accelerate up to 730 rpm., which requires 22 hp. to overcome the friction and represents the balancing speed of the crane bridge of 415 ft. per min.

There are many cases where this high free-running balancing speed will never be obtained due to the short runway. However, on some of the larger mills the crane runway is several hundred feet long and it is easily possible for the cranes to reach the high balancing speed. While these high speeds may not be objectionable, and in fact they may be desirable in many cases for long runs, the fact remains that if the bridge motor is plugged with the motor running at twice its normal speed the torque developed by the motor will be approximately 35 per cent higher than the torque that would be developed if the motor was running at its rated speed. This is due to the fact that the series field of the motor is suddenly strengthened and the counter-voltage instead of being approximately equal to the line voltage is equal to twice the line voltage, which causes more current to circulate through the motor and develop higher than normal plugging torque. It is for this reason that crane wheels will occasionally slip when plugged at high speeds where they will not slip when plugged at normal speeds.

It is not our intention in this short discussion to develop an accurate method for calculating the horsepower requirements of the bridge drive. We do wish to emphasize that the modern crane is geared for considerably higher speed than the older units, and has a better mechanical construction so that the power to overcome friction is proportionally smaller. Therefore, the primary factor in determining the size of the motor is the horsepower required for acceleration and deceleration. Since this is the case any attempt to use a formula combining both of these factors into one constant is apt to give inaccurate results. We feel that any formula to cover all conditions should be made up of two parts, one covering the part to overcome friction and the other covering the part to provide the acceleration and deceleration.

The duty cycle of most shop cranes is of sufficient intermittent nature that the root-mean-square rating of the motor can be more or less neglected and the motor applied on a torque basis using the standard ½ hr. crane rating. However, there are certain cranes that operate on a definite duty cycle performing a given operation all day long. Usually there is a desire to make this cycle as short as possible and the motor is frequently geared for high speeds, the bridge motion sometimes going as high as 600 to 650 ft. per min. In this type of application the root-mean-square rating of the motor is very important and some burned out armatures will be the result, unless a careful duty cycle calculation is made, and the motor frame size selected on the basis of a 5 hour root-mean-square rating instead of its ½ hr. rating.

In order to obtain the high rates of acceleration to meet the duty cycle, it is necessary to increase the accelerating torque of the motor to a point where if the motor is plugged at the high free-running speed there may be sufficient torque developed to cause slipping of the wheels. Therefore, we feel that this duty cycle application is the most critical of all crane bridge applications and requires the proper selection of the motor and control to give satisfactory operation.

The operating condition described above is the exception rather than the rule, and for the ordinary shop crane, the rates of acceleration are sufficiently slow that there is little trouble experienced with slippage of crane wheels. We believe that it is the consensus of opinion that it is better to have these crane bridge drives slightly overmotored to reduce maintenance on the driving motor and then to adjust the accelerating and plugging resistors to give the desired operating conditions.

DETERMINATION OF GEAR RATIO AND MOTOR SIZE

By

J. A. JACKSON

▲ THE duty of a crane bridge motor may be divided into three parts, assuming all retardation to be done by a foot operated brake as is now commonly used:

1. Acceleration while on the resistor.
2. Acceleration on the motor speed torque curve.
3. Full speed running.

The motor fulfills its duty if, without overheating it accelerates the crane while the resistor is cutting out at a rate of acceleration which does not set up excessive hook swing, does not slip the wheels, and does not cause much discomfort to the operator; if, after the resistor is cut out, acceleration on the speed torque curve of the motor does not drag out excessively; if, after attaining constant speed, it propels the crane with rated load on the hook at or slightly above the specified speed; if it has sufficient torque margin to handle extraordinary operating conditions.

This entire discourse is based on the premise that the speed as specified is known to be correct for the service and should not be materially exceeded; that the desired rate of acceleration while cutting out the resistor is reasonably well known and that satisfactory data for determining the friction load on crane bridge drives is available.

The maximum permissible rate of acceleration which occurs while accelerating on the resistor, demands a certain force (fa) parallel to the track at the bridge truck axles.

This force in lb. equals

$$fa = \frac{W \times a}{32.2} \qquad (1)$$

where (W) is the total weight of crane and load in lb. and (a) is the desired rate of acceleration in ft. per sec. per sec. Force (fa) as determined from (1) should be increased by 4 per cent on cranes with small gear reduction and by 7 per cent on cranes with large gear ratios to provide torque for accelerating the armature, brake wheel, gears, shafting, etc. The force (fa) must be further increased by the force (fr) which is the force applied at the truck axles to overcome rolling friction. The sum will give the total force to accelerate the crane. The force (fr) is, with modern cranes, much less than

(fa), hence (fa) largely governs the size of the motor, but (fr) is the controlling factor in determining the gear ratio. The gear ratio, however, can not be determined until the speed of the motor when producing the force (fr) at the truck axle is known, hence a start is made by figuring the horsepower to propel at specified speed (S) or

$$\text{Hp.} = \frac{fr \times S}{33000 \times E} \quad \ldots\ldots (2)$$

where (fr) is the total lb. pull at the truck axles to propel the crane at constant speed, (S) is the specified speed in ft. per min., and (E) is the mechanical efficiency between the motor shaft and truck wheel. Then tentatively select a motor with a half-hour rating about three times the hp. from (2) and from its speed horsepower curve read the speed Rm corresponding to hp. as just determined. The truck wheel speed in rpm. equals

$$Rt = \frac{S}{3.14 \times D} \quad \ldots\ldots (3)$$

where (S) equals the specified speed in ft. per min. and (D) is the truck wheel diameter in feet. Then the total gear ratio

$$G = \frac{Rm}{Rt} \quad \ldots\ldots (4)$$

The total force to accelerate (ft) is the sum of (fa) and (fr) or

$$ft = fa + fr \quad \ldots\ldots (5)$$

This force (ft) can be reduced to torque at the truck wheel by multiplying by the truck wheel radius in feet or

$$Tw = \frac{ft \times D}{2} \quad \ldots\ldots (6)$$

where (Tw) equals torque in ft. lb. at truck wheel; and (D) equals the diameter of truck wheel in feet.

The torque at the motor to accelerate while on the resistor equals

$$Tm = \frac{Tw}{G \times E} \quad \ldots\ldots (7)$$

where (Tw) is torque at the truck wheel; (G) is the total gear ratio from (4) and (E) is the mechanical efficiency of the bridge drive.

If the torque (Tm) so obtained is not over 135 per cent of the ½ hour rated torque of the trial motor, the motor should be satisfactory unless the crane is subjected to a regular heavy duty cycle, in which case the duty cycle must be ascertained and the motor checked for heating. Experience indicates that it is seldom that heating determines the size of a bridge motor on a steel mill crane operating on miscellaneous duty, although it should not be neglected. Cranes operating with magnets, buckets or on a tonnage basis should always be checked for heating.

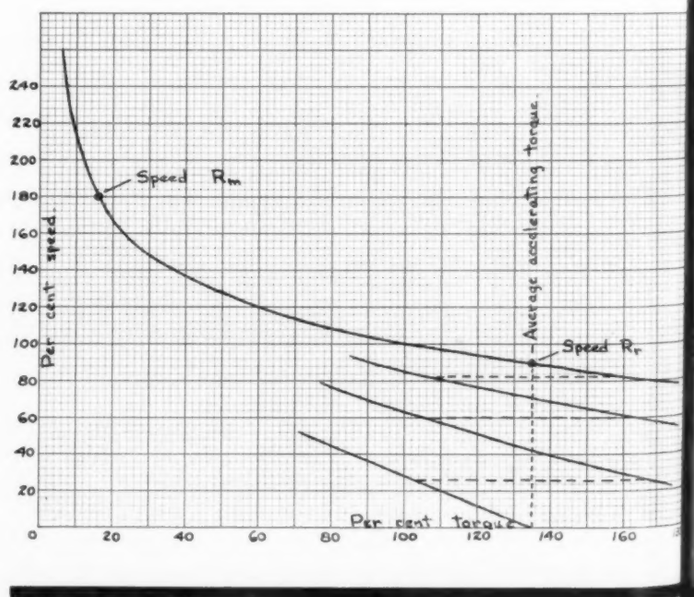
Should the accelerating torque as determined above be greater than 135 per cent of the rated torque of the tentatively selected motor, try the next size larger standard motor. This will require a different gear ratio, as the hp. from formula (2) required to propel at specified speed does not change. From the speed hp. curve of the larger motor, read the speed in rpm. corresponding to the hp. to propel. Then apply formulae (3), (4) and (7) to determine the new gear ratio and the accelerating torque at the motor. Check this against 1.35 times the ½ hour rated torque of the motor.

Seldom will it be necessary to make a third trial.

The rate of acceleration (a) in formula (1) can only be attained while the control is cutting out the resistor. Having determined the accelerating torque (Tm) necessary to produce (a), the resistor taps and timing of the relays can be set to obtain an average current which produces the torque (Tm). The speed of the motor as read from the motor speed curve corresponding to the torque (Tm) will be very much lower (about ⅓ to ½) than the speed (Rm) from which the gear ratio was determined. Call this lower speed (Rr). The crane must accelerate from this lower speed (Rr) to (Rm) on the speed curve of the motor. A study of the speed torque curve (see Figure 1) shows that as the speed increases, between (Rr) and (Rm), the torque falls off rapidly, hence the rate of acceleration decreases rapidly, falling to zero when constant speed is reached at (Rm). At (Rm) all available torque is used for overcoming friction and there is none left for acceleration.

From an inspection of Figure 1, it can be readily seen that the closer together (Rr) and (Rm) are on the speed curve, the shorter will be the time consumed in accelerating on the motor speed curve for, obviously, if (Rm) coincided with (Rr), all acceleration would be on the resistor. Now (Rr) can be moved to the left by adjusting the control to reduce the average accelerating torque while on the resistor. This, however, makes the crane sluggish on initial get-away, and may require more accelerating contactors and relays. (Rm) may be moved to the right on the curve by changing the size of the motor and gear ratio. Reducing the motor size, if permissible, always moves (Rm) to the right along the

FIGURE 1



curve, provided the gearing is correctly selected. This brings (*Rm*) nearer to (*Rr*) and reduces the accelerating time on the speed torque curve, provided the control is set for the same initial rate of acceleration in both cases.

It might be well to point out here that a study of motor size and gear ratio must be combined with control adjustment as affecting initial rate of acceleration if a complete picture is to be obtained.

Perhaps the best way to analyze the differences in operation when using different motors and gear ratios is to calculate the speed time curves from standstill to the point where full speed is reached using several different combinations of motors, gear ratios and control adjustments. The results of such calculations are shown on Figures 2, 3 and 4, and in each case curve A shows the speed time curve determined by test. The right-hand end of each curve shows the point at which the crane reached constant speed. The straight part of each curve starting at zero shows the time the crane was accelerating at a constant rate while the resistor is being cut out. Curves B and C show the effect of changing motor size, gear ratio and initial rate of acceleration or, in the case of curve B for crane No. 8, of changing only the initial rate of acceleration.

Table I shows the results of tests on six cranes, the first, second and fourth of which were used for the curves in Figures 2, 3 and 4. From this it will be noted that the specified speeds of the three cranes are as follows:

Crane No. 10—350 ft. per min.
Crane No. 8—500 ft. per min.
Crane No. 33—350 ft. per min.

The curves show that had motors and gearing been selected to limit the maximum speed to the specified speed or a small amount above as a safety factor, smaller motors would have brought the cranes to their constant speed quicker than they reached constant speed with existing larger motors and the gearing in use. Initial accelerating rates in each case are higher than with the larger motor. On curve B for crane No. 10,

CRANE #10 - SPAN-80 FT - RATED LOAD 10 TONS - WT. 57 TONS (NET) - BRIDGE SPEED 350 FPM - GEARING 15.6

A-45 Hp-15.6 Gearing-0.581 fpsps on resistor-Propelling hp-20% of ½ hr. rating-Accelerating Torque 45%
B-33 Hp-23.5 Gearing-1.2 fpsps on resistor-Propelling hp-23% of ½ hr. rating-Accelerating Torque 89%
C-19 Hp-21.8 Gearing-1.03 fpsps on resistor-Propelling hp-34% of ½ hr. rating-Accelerating Torque 150%

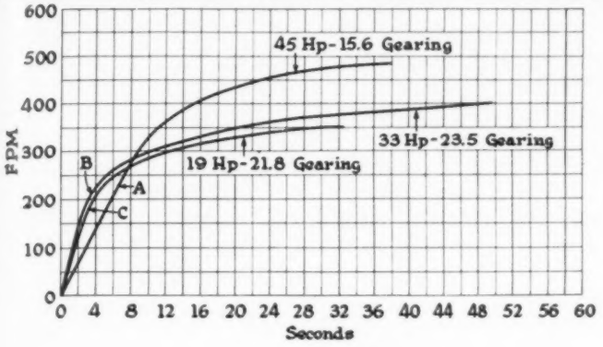


FIGURE 2

FIGURE 3

CRANE #8 - SPAN-90 FT - RATED LOAD 25 TONS - WT. 108 TONS (NET) - BRIDGE SPEED 500 FPM - GEARING 10.1

A-65 Hp-10.1 Gearing-0.624 fpsps on resistor-Propelling hp-26% of ½ hr. rating-Accelerating Torque 89%
B-65 Hp-10.1 Gearing-1.1 fpsps on resistor-Propelling hp-26% of ½ hr. rating-Accelerating Torque 150%
C-45 Hp-10.9 Gearing-0.76 fpsps on resistor-Propelling hp-37% of ½ hr. rating-Accelerating Torque 150%

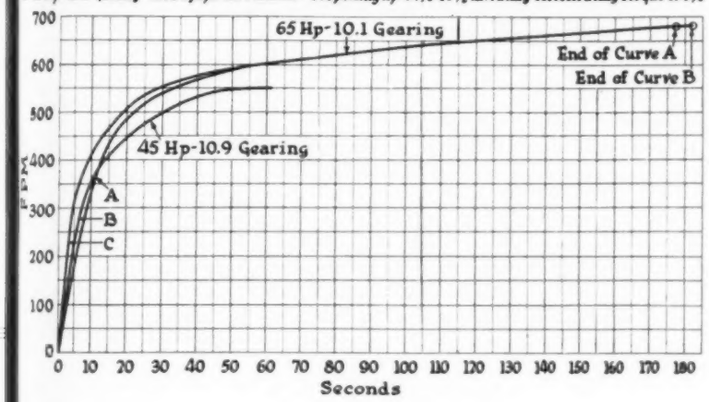


FIGURE 4

CRANE #33 - SPAN-120 FT - RATED LOAD 30 TONS - WT. 160 TONS (NET) - BRIDGE SPEED 350 FPM - GEARING 10.1

A-2-65 Hp-10.1 Gearing-0.7 fpsps on resistor-Propelling hp-16% of ½ hr. rating-Accelerating Torque 76%
B-2-45 Hp-21.1 Gearing-0.81 fpsps on resistor-Propelling hp-23% of ½ hr. rating-Accelerating Torque 66%
C-2-45 Hp-10.1 Gearing-1.0 fpsps on resistor-Propelling hp-37% of ½ hr. rating-Accelerating Torque 150%

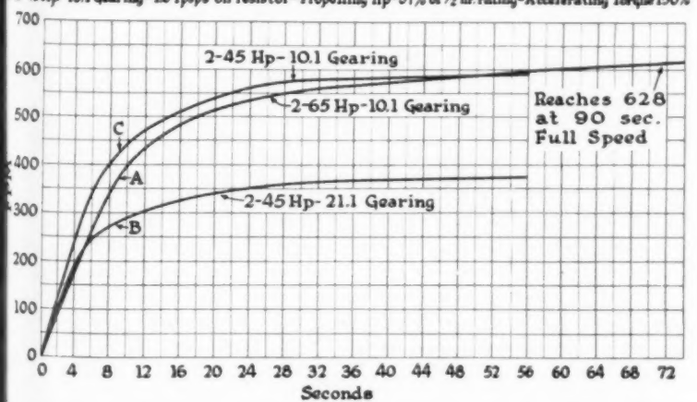


TABLE 1

Crane Number	Net Weight-Tons	Span	Rated Load	Wheel Diameter	Gear Ratio	Motor Rating-30 Min.	Specified FPM	Measured Speed No Load FPM	Measured Speed No Load %	Measured Speed With Load FPM	Measured Speed With Load %	Calculated Speed Rated Load FPM	Calculated Speed Rated Load %	Propelling Torque No Load Lb. Ft.	Propelling Torque No Load % Torque	Propelling Torque Rated Load Lb. Ft.	Propelling Torque Rated Load % Torque	Lb. per Ton-No Load	Lb. per Ton-Rated Load	%Accel. Torque on Resistor	Seconds on Resistor	Sec. to Reach Constant Speed	Connected Hp per Ton with Load	Rated Speed Times Tons per Hp
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
10	57	80	10	24"	15.6	45	350	483	138	—	—	467	133	33.5	6.5	—	—	8.00	—	45	3.75	40	.67	521
8	108	90	25	24"	10.1	65	500	678N	136	—	—	585	117	81.2	10.5	—	—	6.6	—	90	1.5	95	.49	1024
								610S	122	—	—			100	12.9	—	—	8.1	—					
34	48	60	10	24"	15.6	33	350	495N	142	—	—	430	123	44.4	12.8	—	—	12.6	—	79	2.5	37.5	.57	615
								512S	146	—	—	422	122	47.2	13.7	—	—	13.3	—					
33	160	120	30	24"	10.1	2-65	350	652	186	628*	179*	628*	179*	160	10.3	180°	11.6°	8.8	8.45	—	1.0	49	.685	512
39	133	120	15	24"	9.36	2-45	500	637S	127	—	—	627	125	129.6	12.6	—	—	7.93	—	65	5.0	70	.608	823
A	112.5		25	24"	8.04	2-45	500	738	147	711°	142°			160	15.6	190	18.5	10.65	10.9°				.655	764

*27 Ton Load °18 Ton Load

the initial rate of acceleration was made so high that (*Rr*) and (*Rm*) were spread too far apart, causing a long dragout. With a lower rate, but still higher than on curve A, a better showing could be made.

Obviously the larger motors in use could accelerate the cranes at much higher rates, but the fact remains that the lower rates were being used, indicating that they were satisfactory.

It is believed that studies of this kind, togeher with an analysis of the kind of service on which a crane is to be used, will frequently show that with a gear ratio selected to give specified speed and a correctly adjusted control, smaller motors can be used without jeopardizing crane operation.

Cranes doing grab bucket or magnet duty or operating on a tonnage basis subject motors to heavy duty and motor heating generally governs the motor size. Should this result in a size of motor larger than required from a torque standpoint, it is suggested that a motor enclosed on the top half and open on the bottom half be considered as this construction gives greater heating capacity. The use of hydraulic brakes for retarding has reduced the heating duty on motors compared to the heating to which they were subjected when plugging was generally used. This favors the use of smaller motors.

Mill motors will, in emergencies, develop about four times normal torque for half a minute without burning up or damaging the commutator. Since the normal friction load of a crane seldom requires over 33 per cent of the rated motor torque, there is available for short periods twelve times normal constant speed requirements. As for accelerating requirements, the average available torque for normal accelerating rate as determined by the control adjustments will move a crane handling an extra heavy load (provided the crane is not jammed), the rate of acceleration being reduced in the same ratio that the mass is increased. In this connection it should be remembered that double rated load on the hook adds only a relatively small amount to the total mass. In the six cranes tested the average increase in mass is 16 per cent with a maximum of 19 per cent. In all probability the timing of the relays will, under the heavy load condition, allow the current peaks to pile up and maintain a rate of acceleration almost up to normal. It is doubtful if any other part of a crane has any greater factor of safety for overloads or any higher degree of flexibility with safety.

From the standpoint of sound engineering and economics, the total overall cost of the gearing and motor should be a minimum. Electrical equipment costs more per pound than gearing due to the high price of copper and magnetic steel plus the intricate manufacturing processes required on electrical equipment, hence a smaller motor and a correct gear ratio should produce the lowest price. The fallacy in using an oversize motor is that, in addition to a higher price motor, the gearing will cost more as well if it is correctly selected to maintain the specified full speed of the crane. This can be readily understood by recognizing that the horsepower to propel at specified speed is constant regardless of the motor used, and a larger motor, due to its series characteristic, will be running at a lower percentage of its rating; hence it will be rotating at a higher rpm. than a smaller motor. Thus for the same ft. per min. of the bridge, the gear ratio must be larger. Furthermore, if the gearing is built to safely stand the permissible overload torque which the larger motor can produce, another increase in gearing cost may be introduced. Now let's assume that the larger motor is used, but with the same gear ratio that would give correct specified speed with a smaller motor of ample size. Then the specified ft. per min. will be exceeded and as the kinetic energy in the crane increases as the speed squared, a larger, more expensive brake is required to stop the crane in a recognized safe distance. Also many of the stresses on the crane runways and columns increase as the speed squared. Unless designed for such stresses, building maintenance may be increased. If a brake designed only for the specified speed is used, the higher speed may jeopardize the safety of the crane and building.

Experience in any mill operation is an excellent asset, but when the time comes to make an economic study to determine whether the operation of a certain machine justifies its initial capital investment, a mathematical investigation is an invaluable aid to experience and may reveal facts which experience overlooks.

COMPOUND WINDING FOR CRANE BRIDGE MOTORS

By

J. R. LEWIS

▲ FOR more than 35 years we have been building direct current mill type motors for application to cranes, and we make a safe venture in saying that in that long period, the majority of such d-c. mill type motors have been supplied with series winding. Due to the excellent progress made by the control manufacturers, the selection of control—whether the application be bridge, trolley or hoist—has been such that very little difficulty has been experienced with series wound motors.

However, a new thought has now been injected at this meeting, and that is the idea of direct current mill type motors for crane bridge application being furnished with compound field winding, instead of straight series winding. To me, it would seem the use of compound winding would be quite advantageous for direct current mill type motors on crane bridge travel. It is also my private opinion that we figure too liberally on the size of mill type motors—whether same be used on bridge, or trolley, or on hoist. Therefore, with too much horsepower in the motor, one would often be tempted to let such a hoist, or bridge on a crane overrun its regular distance and consequently, there would be a decided advantage to have some shunt winding in our mill type motor.

Many of the motor manufacturers have more or less adopted for "compound winding," the idea of approximately 50 per cent shunt and 50 per cent series winding. We do not care whether the steel mill uses this ratio of 50 per cent shunt and 50 per cent series for compound winding in mill motors, but I would like to see a general

practice of using what we formerly called the term, "series shunt," instead of the term "compound wound" for such motors. In any event when the steel mill has in mind the use of mill type motors, it would seem to me as if the user could well specify some percentage of shunt winding in their mill motors whenever it is their intention to use same on a crane for bridge application.

Mr. R. J. Harry has mentioned in his paper that generally speaking the running speed of the bridge on a crane is often on the average of 2½ times the speed of the motor. This means that we often see the running speed of the bridge on steel mill cranes operate as high as 1200 ft. per min. You can well see, therefore, that having some shunt winding in the fields of the motor for such a bridge application to a crane would be a decided advantage.

Of course, we realize that the two main features to be considered on bridge application to a crane are: first, acceleration, and second, the proper gear ratio.

Anyone that gets an idea you can use any kind of small motor on the bridge of a crane will be horribly fooled. You must get the proper acceleration and you must have the proper gear ratio.

Therefore, as long as the bridge application and its operating characteristics have been taken care of, we see no reason for the steel mills not adopting a more and more coming tendency to specify series shunt winding in these direct current mill type motors. We know that the control manufacturers have already developed the proper design of control for either compound or straight series mill type motors, but even at that we lean quite favorably towards the advantage of using compound winding for mill type motors.

RESISTOR VALUES AND ACCELERATION

By

P. B. HARWOOD

▲ ALTHOUGH the calculation of the horsepower required to accelerate a crane bridge is not properly a part of this paper, the following discussion is given as a matter of interest and because it ties in somewhat with the resistor design. Each crane manufacturer undoubtedly has his own method of determining the proper horsepower. The method discussed here has been used with success as a basis for controller design. The equation for the average accelerating horsepower is:

$$\text{Hp}_a = \left[\frac{W \times A}{32} + F\right] \frac{S}{\text{Eff.} \times 33000}$$

In this equation:

W = the weight of the bridge in pounds.

A = the rate of acceleration in feet per second per second.

F = the friction force.

S = lineal speed of the bridge at full load motor speed.

Eff. = the efficiency of the gearing.

Average values for sleeve bearing cranes may be as follows:

A = 2 ft. per sec. per sec.

F = 20 lb. per ton, or 1 per cent of W.

The bracketed portion of the equation represents the force required to move the bridge in the direction of travel—the expression $W \times A/32$ being the force required for acceleration, and F being the force to overcome friction. When the motor has been accelerated to its normal speed and at that instant is running at 100 per cent speed and drawing 100 per cent current, then the full normal horsepower of the motor is being delivered and applied to the load. During the accelerating period the same horsepower is required of the motor, but only a portion of it is being delivered in useful work, and the remainder is being dissipated in the control resistor.

This method of calculation would indicate that since F is 1 per cent of W, or 16 per cent of $W \times A/32$, complete elimination of the friction would permit a reduction of 16 per cent in the motor rating. Partial elimination of friction by the use of anti-friction bearings would reduce the motor horsepower by something less than 16 per cent.

In actual practice the control manufacturer usually has nothing to do with the motor selection but is simply advised of the frame size and rating for which the control must be designed. His problem is to design a suitable resistor of the proper ohmic value, number of steps, taper, and current carrying capacity for both acceleration and plugging service. In any calculations involving a series motor, consideration must be given to the fact that the load current flows through the motor field, and the strength of the field therefore varies with that current, but not in direct proportion. The torque under any condition is calculated from the equation:

$$T = I_a \times F_s$$

in which I_a is the armature current and F_s is the value of the field strength or flux. To determine the value of F_s a curve must be drawn to show the relation of the field current and the field strength. Such a curve is shown in Figure 1. For very accurate results the curve of the specific motor in question should be used, but the design characteristics of mill motors are such that the flux curves for various sizes and various makes will be found to be very much the same. The curve of Figure 1 has been checked against the actual curves of a number of motors and found to be close enough for any practical purpose.

The speed under any condition may be calculated from the equation:

$$S = \frac{E - RI}{F_s(E - R_a I)}$$

In this equation E = the line voltage, R = the total resistance in circuit, I = the current, F_s = the field strength, R_a = the armature resistance. Inspection of this equation will show that the expression $E - RI$ is the counter-voltage of the motor. The expression $E - R_a I$ is the

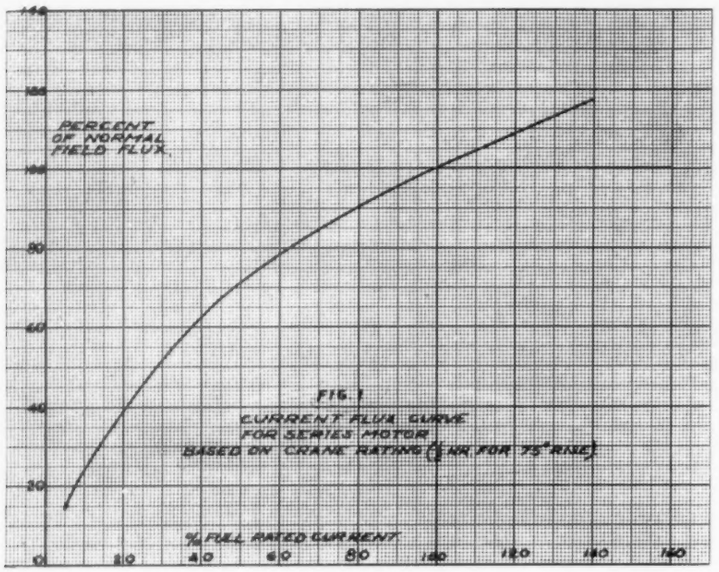

PERCENT OF NORMAL FIELD FLUX
FIG. 1
CURRENT FLUX CURVE FOR SERIES MOTOR BASED ON CRANE RATING (½ HR. FOR 75° RISE)
% FULL RATED CURRENT

FIGURE 1—Current flux curve for series motor, based on crane rating (½ hr. for 75 degrees rise).

countervoltage of the motor when it is running at full load and full speed, or, in other words, is the line voltage minus the normal armature drop. A fair value for the armature drop, which actually includes the drop in the field and series brake, and in the connecting leads, is 11 per cent.

From the equations given for torque and for speed it is possible to plot a set of speed torque curves provided the resistance in circuit on the various control points is known. If an attempt is made to approach the control design problem by first drawing a set of desirable speed torque curves and then calculating the control resistor, the calculations become quite involved. In order to simplify the discussion and the calculation, this paper has assumed a set of resistor values as actually used in practice, and from those values the speed torque curves of Figure 2 and Figure 3 have been drawn. Figure 2 shows the curves for a controller using one plugging contactor and three accelerating contactors, and Figure 3 shows the curves for a controller using one plugging contactor and four accelerating contactors.

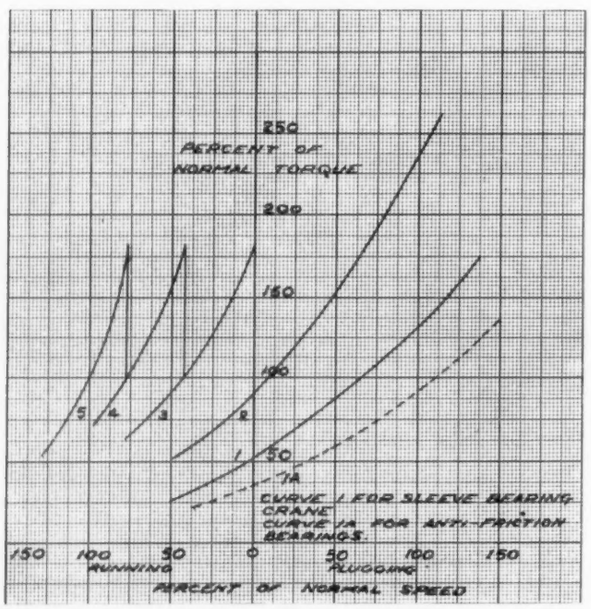
The NEMA standard ratings for crane controllers are:

8-hour contactor rating, amp.	Crane rating, amp.	Hp.	Minimum number of accelerating contactors, not including plugging contactor
100	133	35	3
150	200	55	3
300	400	110	3
600	800	225	4

The speed torque curves of Figure 2 are obtained with a resistor designed as follows:

Step	Ohms in per cent of E/I for sleeve bearings	Ohms in per cent of E/I for anti-friction bearings
1	50	92
2	40	40
3	30	30
4	25	25
Control resistance	145	187
Motor resistance	11	11
Total	156	198

FIGURE 2—Speed torque curves of crane motor, one plugging contactor, three accelerating contactors.



Starting from rest, and moving the controller to the first point, the inrush current will be,

$$I_1 = \frac{E}{1.56\,\frac{E}{I}}$$

$$= 0.64I$$

With this current the field strength, from Figure 1, will be .81. The torque will be,

$$T_1 = I_1 \times F_s$$

$$= 0.64 \times 0.81$$

$$= 0.52 \text{ or } 52 \text{ per cent of normal.}$$

Therefore, curve 1 crosses the zero speed line at 52 per cent torque. Any other point on the curve may be determined by assuming a value for the current and

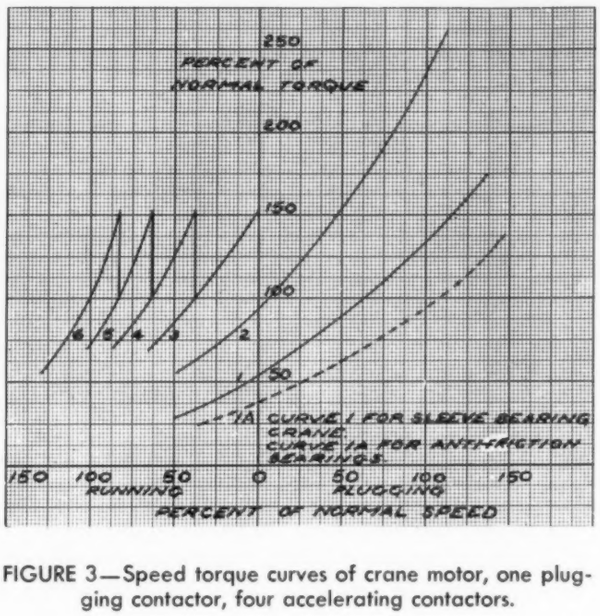


FIGURE 3—Speed torque curves of crane motor, one plugging contactor, four accelerating contactors.

calculating speed and torque. For example, assume

$I = 1$ (100 per cent), then

$F_s = 1$

$\text{Torque} = I \times F_s = 1$

$$\text{Speed} = \frac{E - RI}{F_s(E - 0.11)}$$

$$= \frac{1 - 1.56}{1(1 - 0.11)}$$

$$= -\frac{0.56}{0.89}$$

$= -0.63$ or 63 per cent in the reverse direction. The curve therefore crosses the 100 per cent torque line, at 63 per cent reverse speed. Similarly, any other points may be determined.

When the controller is moved to the second point, step 1 is short circuited, and the total resistance is then 1.06. Curve 2 is plotted using that value, and in a similar manner the rest of the curves are drawn to show the conditions on each point of the controller.

A study of the curves of Figure 2 will show that if the bridge does not move on the first control point, with 52 per cent of torque applied, the torque obtained on the second point will be 91 per cent. If the bridge still does not move, the third control point will apply 183 per cent torque. If the bridge now starts, and if the control is properly set, acceleration will proceed along curves 3, 4, and 5, with maximum peak torques of 183 per cent, and minimum applied torques of 100 per cent. In practice, the peak torque will not be as high as the theoretical value, because the motor field will not build up instantaneously when the current is increased. Oscillographs show that the vertical lines of Figure 2 will actually slant to the left, that is, the motor will accelerate a certain amount during the build-up of current. The sharp peaks appear rounded off, and where Figure 2 shows a peak of 183 per cent, an oscillograph will show an actual peak of about 165 per cent.

Curve 5 is that of the motor across the line without external resistance. Since the motor has been selected to have the necessary torque to accelerate the bridge at the desired rate, it will have more than enough torque to keep the bridge moving, and consequently will continue to accelerate beyond full rated speed. The ultimate speed reached will depend on the friction losses, and on the length of the bridge travel. Curve 1 shows that if the motor reaches full speed, and is plugged at that point, the torque obtained will be 132 per cent. If the motor reaches a speed of 140 per cent, a plugging torque of 183 per cent, equal to that of acceleration, will be obtained. These values have been found satisfactory for sleeve bearing cranes, but with anti-friction bearings the over speed may be much higher, and the plugging torque would be too great. Consequently, controllers for such cranes have the ohmic value of the plugging step increased from 50 per cent to 92 per cent of E/I. Curve 1A shows the results obtained with that arrangement. The starting torque on point 1 is reduced to 36 per cent, the plugging torque at full speed to 93 per cent, and the torque at 50 per cent over speed is 140 per cent. These values will give good braking without slipping the wheels.

Figure 3 shows the results obtained when an additional accelerating contactor is used. The resistor design is then,

Step	Ohms in per cent of E/I for sleeve bearings	Ohms in per cent of E/I for anti-friction bearings
1	50	92
2	30	30
3	24	24
4	22	22
5	19	19
Control resistance....	145	187
Motor resistance.....	11	11
Total.....	156	198

Comparison with Figure 2 will show that the total ohms are the same, and also the first steps. Therefore, curves 1, 1A and 2 are unchanged. The plugging conditions are the same. The only change is an increase in the number of accelerating steps, with corresponding reduction of the maximum accelerating peak torque from 183 per cent to 151 per cent. The practical advantage is a somewhat more smooth acceleration, because the torque is applied in smaller increments.

With magnetic control, acceleration as shown by these curves is obtained regardless of the number of master points. A multi-point master is universally used, however, to obtain speed control. It is possible to get one more speed point than the number of resistor contactors, that is, five speeds for Figure 2, and six for Figure 3. Since the majority of cranes fall in sizes which use a total of four contactors, the use of five speed points appears to be common practice for all sizes. The speed change between points 5 and 6 is

relatively small, and these contactors are commonly controlled on the same master point.

The initial inrush current with the sleeve bearing design is 64 per cent, and with the anti-friction bearing design is 50 per cent. The recommended resistor capacity, based on these inrush values, will be

For light duty sleeve bearing cranes..NEMA Class 153
For heavy duty sleeve bearing cranes.NEMA Class 163
For light duty anti-friction bearing cranes.....NEMA Class 152
For heavy duty anti-friction bearing cranes.....NEMA Class 162

TORQUE REQUIREMENTS FOR CRANE BRIDGE BRAKING

By

R. L. PUETTE

▲ THE general subject of torque requirements for crane bridge braking is extremely difficult to condense to any group of formulae due to the number of unknown values and the variations that exist between crane installations.

The maximum braking torque is determined and limited by the slipping of the wheels on the crane rails. The value of maximum torque can be estimated by knowing the following values:

(A) Weight on driven wheels of the crane with and without load.
(B) The number of driven wheels.
(C) Condition of the rails.
(D) The rate of travel of the crane bridge.

It is not difficult to determine the values of (A) and (B) in the above tabulation. Item (C) covering the condition of the rails themselves, of course, determines the coefficient of friction that will exist between the wheel or wheels and the rail. A tremendous variation in the value of the coefficient of friction exists and for this one reason it is almost impossible to calculate the maximum torque that can be applied for stopping the crane without slipping the wheels. The coefficient of friction will vary to some extent with the variation in speed of the crane bridge. The coefficient of sliding friction will vary between .25 for a clean dry rail to .015 for a wet rail and much less if oil or grease is present.

It has been fairly well demonstrated that approximately twice the full load motor torque will slip the wheels of a crane during acceleration. This value is only approximate and of course assumes that the motor has been properly selected in terms of its horsepower and torque rating for the particular crane. We could, therefore, estimate that on average conditions with a properly selected motor, and dry rails, that the bridge wheels would be slipped during braking if we applied approximately twice the full load motor torque in the form of a braking torque.

The speed, inertia, and distance permitted for stopping will determine the average torque needed, providing, of course, we do not exceed a value where the crane wheels will slip. An old rule of thumb was that a crane bridge should be capable of being stopped by the bridge brake in approximately 10 to 15 per cent of the maximum speed of the crane in feet per minute. That is to say, a crane bridge traveling at 300 ft. per min. was considered with a view toward being able to be stopped in approximately 30 to 40 ft. Today crane bridge speeds have gone up to values in the order of 500 or 600 ft. per min. and observation indicates that they are still being stopped, where required, in about the same distance as of old, namely, about 30 ft. Obviously the bridge brake is having a considerably harder job to do since the inertia varies as the square of the rate of speed.

If we wish to calculate the torque required to stop a crane in a given distance, the following formula can be used, if wheel slippage does not occur.

$$T = \frac{WR^2 \times N}{308 \times t}$$

where T = torque in pounds feet at the motor shaft.
WR^2 = moment of inertia of the entire crane including motor armature referred to the motor shaft.
N = change in speed, which for complete stop is equal to motor speed in rpm. at the speed which the crane is running.
t = time in seconds to stop.

Ordinarily the bridge brake is applied on the basis of full load motor torque, which can be calculated as follows:

$$T = \frac{\text{hp.} \times 5250}{\text{rpm.}}$$

where T = torque in pounds feet at motor shaft.
hp. = horsepower rating of motor.
rpm. = full load speed of motor.

Some crane manufacturers insert a service factor in the enumerator of the above formula which varies from .75 to 1.5 to determine bridge brake size. This service factor is based on the duty for the crane and for steel mills is 1.25 to 1.5.

If we assume that the crane wheels on a given installation will not slip and determine the value of torque required to stop the crane in a given distance, we are still faced with the problem of knowing accurately the torque rating of the brake itself. If we consider, for a moment, the value of torque produced by a magnet operated brake, we know that the value of torque is determined by the pull of the magnet coil and the spring. There is a definite maximum value for any given size brake and this is definitely known by each individual manufacturer. Now considering the bridge brake, whether it be operated by mechanical linkages or hydraulic means, the value of torque produced as a maximum is determined by the weight of the crane operator. There is no standardization of weight of crane operators.

Another factor governing the size of the bridge brake is the type of control apparatus handling the motor and the use that the crane operator makes of this control. Present day magnetic control provides controlled plugging values and if the operator, when stopping the crane, uses this retarding torque as a braking action, the service on the mechanical bridge brake is very much reduced and the brake then need only be

used as a spotting device. This does not mean that the bridge brake should be selected small for this reason alone, but it is a means of reducing maintenance on the bridge brake. The motor, if properly selected for the particular drive, should stand without overheating the plugging current imposed. Control apparatus and resistors are on modern magnetic control equipment designed for this type of service.

There are available on the market today mechanically operated or hydraulically operated bridge brakes which are rated by the manufacturers in the maximum torque that the particular manufacturer believes is reasonable for the brake based on reasonable wear of lining and general maintenance.

Generally speaking, the only guide for applying and selecting a bridge brake size is the motor horsepower and speed and this in general is a rather poor indication of the requirements without knowing details of the particular crane involved. Brakes have been in general, applied on the basis of their rating producing approximately full load motor torque, and where this particular value becomes close to the maximum indicated for a given size of brake, the next size has usually been selected. On average conditions this is worked out very well, although a great many variations are possible and the brakes should be liberally rated to stand the maximum service with a minimum lining wear and maintenance cost.

There seems to be a question as to whether bridge motors are not too large and if application practice reduces the size of bridge motors probably a revision in bridge brake application will also have to be modified to avoid getting too small brakes on bridge.

The bridge brake can be considered as a means of dissipating energy in the form of heat. If a brake is selected whose torque rating is capable of sliding the wheels under maximum conditions and due to the duty cycle the brake gets too hot, causing excessive maintenance, the solution is not a larger brake but effective means to get rid of the heat generated more rapidly. Since most of the heat is dissipated by the wheel it is possible to increase the length of the wheel or use fins or similar means to increase the radiating ability of the unit.

This subject is a good one for further investigation which can be materially aided by the users of cranes tabulating the crane data of weight, speed, horsepower, duty, versus the size and maintenance of the bridge brake. Probably for reasons of the unknown factors involved, bridge brakes have, in general, been larger than necessary, although specific instances come to mind where exactly the reverse is true.

PLUGGING AND DYNAMIC BRAKING CRANE BRIDGE BRAKES

By

H. L. WILCOX

▲ THIS paper is limited to the discussion of plugging and dynamic braking of bridge drives together with resistor values for the same. With slight modifications, it would apply to trolley drives as well, especially those on man-trolley cranes but for the sake of clarity let us consider only the bridge motion of mill cranes where the operator's cab is carried on the bridge.

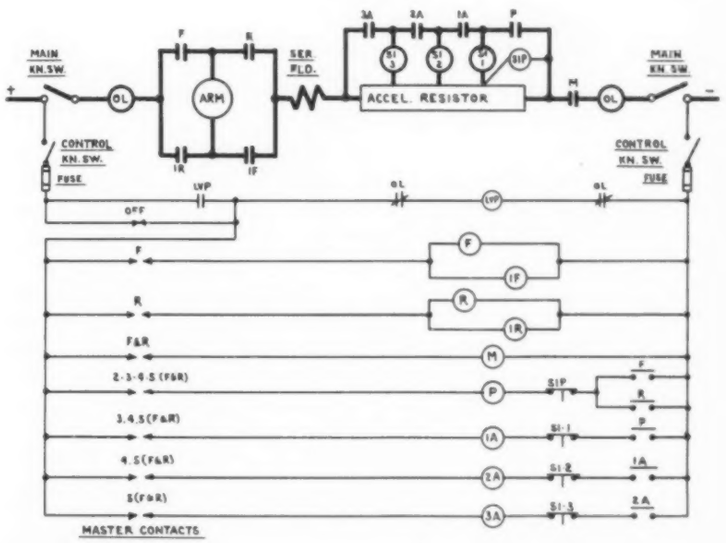
This control is known as reversing plugging magnetic control for bridge drives. Such a controller may be furnished in two ways: (1) with protection; (2) without protection. These are defined by NEMA in Industrial Control Standards for 1937, IC-14-12.

The simplified wiring diagram of a reversing plugging controller with protection for a single motor is shown in Figure 1. It consists of four single pole reversing contactors, accelerating relays and contactors, an accelerating and plugging resistor, a plugging relay and contactor, a double pole fused control circuit switch, a double pole main line knife switch with padlock clip, two automatically reset magnetic overload relays, a negative line contactor and an under-voltage relay.

The simpler form corresponding to "without protection" consists of the same elements as for "with protection" except omitting the double pole main line knife switch with padlock clip, the two automatically reset magnetic overload relays, the negative line contactor and the undervoltage relay. This form is used where the protective devices of all the motions are assembled into a common crane protective panel. Acceleration is being covered by other papers of this symposium so I will pass directly to reversing and plugging.

In many crane movements, stopping is merely preliminary to starting reverse. It is therefore convenient to be able to throw the master handle from forward full over to reverse, causing the bridge to slow down gradu-

FIGURE 1—Reversing plugging control for crane bridge or trolley, with protection.



ally to standstill and then accelerate normally in the reverse direction. For stopping, the operation is the same except the master is brought to the "off" point as the bridge comes to rest.

This plugging to reverse or plugging to stop must be automatic and of such a nature that it will not abuse the motor or throw the operator out of his cab. The ideal conditions would provide a decelerating torque which is equal to the accelerating torque when starting from rest. If the deceleration and acceleration is carried out smoothly and without interruption, the operator will not feel the exact instant of standstill.

In order to provide a decelerating torque under plugging which is equal to the accelerating torque from standstill, it is necessary to provide additional resistance in series with the motor, known as the plugging resistance, in order to hold the current on plugging equal to the amount taken by the motor at starting from rest. Since the torque developed by a series motor increases almost as the square of the current, we cannot permit large plugging currents or it will cause undue shock on the crane and most likely cause the wheels to slip. Once the wheels start to spin it is possible for the motor to accelerate to full speed reverse with the wheels spinning, while the bridge is still sliding along in the original direction.

The amount of plugging resistance required depends somewhat upon the weight of the bridge structure, whether the bridge is over-motored or not, and whether equipped with sleeve or roller bearings. If the bridge is equipped with sleeve bearings and not over-motored, it is usual to provide a resistor of class 153-P or 163-P which allows approximately 70 per cent full load current at standstill and twice as much on plugging. If the crane is equipped with anti-friction bearings, the bridge will attain higher speeds and therefore when plugged will draw heavier currents resulting in higher retarding torques. It is therefore better for roller bearing cranes to supply resistors of class 152-P or 162-P allowing 50 per cent full load current at standstill.

While the 152 and 162 classification allows 50 per cent full load inrush and the 153 and 163 class allows 70 per cent inrush, they are all designed so that when the plugging contactor closes at standstill they will give a first point acceleration peak of 150 per cent full load. The 150 class is good for one 15 second start each 60 seconds or ¼ time on while the 160 class is good for one 15 second start each 45 seconds or ⅓ time on.

Many times, in the interest of duplication of motors, certain motions are greatly over-motored. Notice of these conditions, where they exist, should be transmitted to the control builder so that adequate allowance in resistors may be made. Failure to do so will inevitably result in too low resistance values and unsatisfactory acceleration and plugging.

The plugging relay used to prevent the closure of the plugging contactor until the motor comes to standstill has a real function to perform. If there were no variables such as varying voltage, change in resistance with heat, and changes in load, it would be a relatively simple matter to provide a relay which would operate at the instant of standstill when the motor was being plugged and would close immediately when starting from rest.

The plugging relay may be a current relay connected in series with the armature or a voltage relay connected across the armature and a portion of the resistor connected across resistance only. The voltage relays are more generally used for this purpose. The relay will have either normally open or normally closed contacts. If the former, it must operate on a rising voltage and the only place that this can be found, during plugging, is across the adjacently connected armature and plugging resistor. Since the generated-emf. in the armature is in the same direction as the drop over the resistor, the voltage across the two is the difference of the components which starts at a low value and increases as the motor comes to rest. If this relay is to operate at standstill with 85 per cent normal voltage on the system, which it must or otherwise hold up acceleration, it will operate prematurely for normal or high voltage and this is in a direction contrary to the best interests of the motor and the application.

The type SIP relay shown in Figure 1 is a shunt wound magnetically operated relay with special magnetic circuit which provides a dropout voltage very close to the pick-up voltage of the relay. The relay coil is connected across the plugging resistor R1-R2 which is ½ of the total resistance, and therefore when starting from rest, is subjected to ½ of line voltage. The relay has normally closed contacts in series with the plugging contactor, and is adjusted so that it will not pick up on ½ normal voltage. It therefore allows the plugging contactor to close immediately when starting from rest. When the motor is being plugged, however, the relay coil is subjected to a voltage which is equal to ½ the sum of the line voltage and the counter-emf. and therefore picks up quickly to prevent the closure of the plugging contactor. When the motor comes to rest under plugging, the voltage across the coil, having reached the value for which the relay is set, causes the contacts to close energizing the plugging contactor.

The connection used allows this relay to have normally closed contacts which means that if the voltage across the relay coil is less than ½ voltage, the relay contacts will be closed and the plugging contactor will close. There is no chance, therefore, of the relay holding up operation because of low line voltage. With variations in applied voltage the operation of the relay varies in the right direction. Assuming the relay has been set to bring in the plugging contactor exactly at standstill with normal voltage, then with high voltage it will hold in a little after the motor has come to standstill and has reversed, while on low voltage it will permit the plugging contactor to close just before the motor reaches standstill. This allows approximately the same current to the motor when contactor P closes whether the voltage is high or low.

Another feature in favor of the connection as shown in Figure 1 is that the relay coil is subjected to voltage only while the motor is being plugged. As soon as contactor P closes, the relay coil is shorted out and therefore is not being heated up during the operation of the motor as would be the case with a relay coil connected across the armature. The relay coil therefore maintains a more uniform temperature and allows a more definite operation.

If the resistors are of a material whose resistance increases with temperature, then further allowance

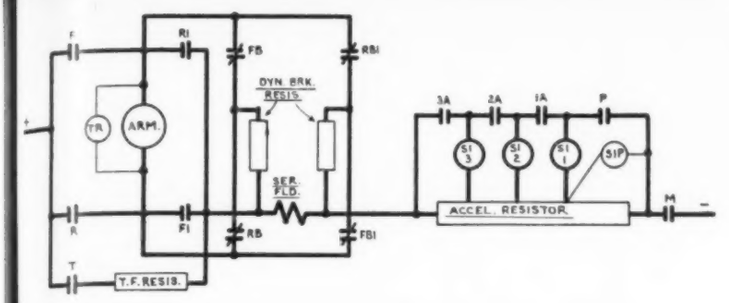


FIGURE 2—Reversing plugging control with safety dynamic braking, without protection (main circuits only).

must be made in a relay connected across the armature since the voltage applied to its coil will decrease as the resistance of the resistor increases. With a relay as connected in Figure 1, the relay coil will always get ½ voltage at standstill, irrespective of changes in resistance with temperature because the coil is connected across ½ of the resistor.

There is still one safety feature which has not been covered, and that is the ability to stop quickly after a failure of power. An operator may find himself approaching the end of travel at a high rate of speed without realizing that power has failed. When he throws his master to reverse for slow down, the bridge continues on unchecked. He cannot stop it. When he realizes what has happened, he jumps on his foot brake if he has one, but may be too late. If it is a man-trolley crane, he cannot have a foot brake on the bridge drive so his only means of stopping are the ominous bumpers at the end. The solution to this problem is the safety dynamic braking stop for bridge drives which provides automatic stopping by dynamic braking the instant power fails.

The simplified main circuit diagram of the safety dynamic braking stop used with a reversing plugging controller for bridge is shown in Figure 2. The spring-closed contactors FB and FB1, RB and RB1 operate in pairs. With power on, all of these spring-closed contacts are open. (When the master is moved to "forward" both pairs of contactors are held open by power, but in addition RB and RB1 are latched open so that they cannot close on failure of power.) A directional switch operates the latch from one pair of contactors to the other for each change of direction. The relay TR closes at a low value of armature voltage to energize the teaser field contactor T and holds in to a very low value. On failure of power in the forward direction, the spring-closed contactors FB and FB1 will close to form a dynamic braking circuit in such a way that the dynamic braking current strengthens the series field of the motor. On failure of power in the reverse direction, the spring-closed contactors RB and RB1 complete the dynamic braking circuit.

During normal operation of the motor on power, the teaser field circuit constitutes a mild armature shunt on the motor. This prevents excessive speeds on long runs and is a very desirable feature especially on drives equipped with anti-friction bearings.

The combination of reversing plugging controllers on series motors operating bridge and trolley drives together with the safety dynamic braking panel, constitutes the ultimate in safety for crane bridge and trolley drives.

CRANE BRIDGE BRAKES

By

J. C. COX

▲ THE fitting of efficient, powerful brakes to crane bridge drives is now a practical necessity. Higher bridge speeds have made plugging an undesirable practice; and from automobile experience, the operator expects reliable action from the bridge brake. Damage to buildings and crane equipment are minimized by good brakes as are road collisions between motor cars. Likewise, better brakes mean faster schedules, whether it be delivering the meat, or loading a freight car.

Assuming the foregoing to be true, let us investigate the problem facing the engineers in charge of the bridge brake problem. Conditions to be dealt with are:

1. Minimum stopping distance necessary with maximum load.
2. Available friction between track wheels and rail.
3. Number of wheels requiring brakes.
4. Strength of gearing through which brakes are operating.
5. Location of brake wheel.
6. Brake size.
7. Work required to operate brakes.
8. Most efficient means for actuation of brakes.

It is evident that the weight of the loaded crane has a considerable bearing on stopping distances—large loads customarily being handled at a slower rate. But certainly a safe, quick stop should always be available in emergencies. It has ofttimes arbitrarily been stated that ten per cent of the bridge speed in feet per minute is the maximum stopping distance. There are two excellent reasons why this is not the best way to write a specification—first, stopping distance is a function of the square of the velocity; hence the formula is misleading, both theoretically and from the operator's practical experience. To stop in 125 ft. from an initial speed of 1250 ft. per min. simply fails to create a feeling of safety to a man who drives an automobile. The second is that operating conditions should influence requirements and consideration be given to "how fast must the crane stop to be safe?"

The theoretical miuimum braking distance is dependent on the rail friction. To attain the maximum rate it is necessary to quickly apply the full braking effort necessary to just avoid sliding the wheels and to maintain this condition throughout the stop. With good brakes we may assume full braking power, and look to

the rail friction to see what braking rate may be expected.

If the rail friction is 0.2 the maximum retarding force at the wheel is .2×wheel load, and since retarding force = mass × acceleration,

$$\text{Acceleration} = \frac{\text{retarding force}}{\text{mass}} = \frac{\text{retarding force} \times 32.2}{\text{wheel load}}$$

in feet per second per second if retarding force and wheel load are given in pounds.

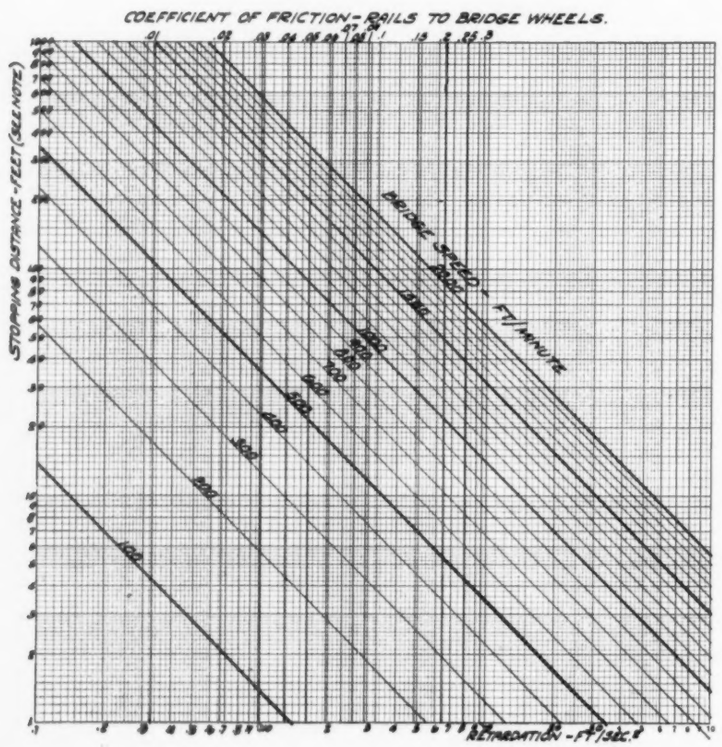
The formula given above holds only if all wheels are braked. More usually, brakes are applied to but two of four wheels, in which case the maximum retardation rate will be but half of the theoretical, provided the wheels are equally loaded. It seems safe to assume that all wheels will carry nearly equal loads, hence in general:

$$\text{Retardation} = \frac{\text{No. of braked wheels} \times K \times 32.2}{\text{total number of wheels}}$$

in feet per sec. per sec.) As the maximum desired rate cannot be expected to equal the theoretical limit, and further because weight distribution during the braking period may not permit equal wheel loading, border-line cases should be handled by fitting an extra set of brakes to the free wheels.

It is essential also that consideration be given to the possibility of decreasing rail friction, due to ice, water, or grease. Where impaired friction is expected, it may be necessary to utilize all available wheel load by braking every axle. In any event, to brake all wheels gives the best theoretical performance. Figure 1 is a convenient chart for obtaining stopping distances, etc.

FIGURE 1—Chart showing stopping distance under various conditions. (Note: Stopping distance corresponds to retardation—ft. sec.- axis. Where all wheels are braked, minimum stopping distance can be estimated if wheel friction is known. It is reasonable to assume a rail friction value between .1 and .2 in the absence of actual data. Where brakes are not applied to all wheels stopping distances can be estimated by finding retardation value on chart corresponding to friction coefficient then multiplying retardation by per cent of total load on braked wheels, then finding stopping distance corresponding to this value.)


COEFFICIENT OF FRICTION - RAILS TO BRIDGE WHEELS.
STOPPING DISTANCE - FEET (SELENOID)
RETARDATION - FT/SEC.²
PERIPHERAL SPEED - FT/MINUTE

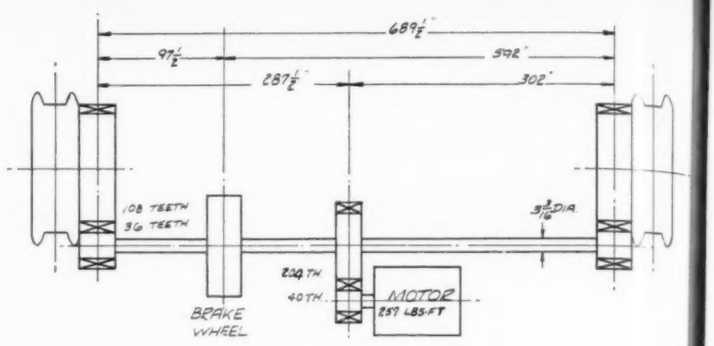


FIGURE 2—Shaft deflection can be a serious matter if brake wheel or motor pinion is considerably off-center.

As it is customary to brake through the gearing used to drive the bridge, pinions, keys, shafting, and associated parts must be designed with ample safety factors. Bearing in mind that braking stresses are of opposite sense to the accelerating forces, and both are quite suddenly applied, fatigue failure must be guarded against by generous proportioning.

It seems obvious that the brake wheel should be in the center of the drive shaft if not on the motor shaft extension. Shaft deflection can be a serious matter if the motor pinion or brake wheel is considerably off center, even though relatively large shafts are used.

Brake size has varied through wide limits on cranes of various designs. In some instances, brake wheels of potentially great capacity are fitted with shoes, which although large, are inefficiently constructed; in which case a smaller brake could have been used, with a resultant lower cost and reduction in rubbing velocity. Brake friction materials have always been the center about which the mechanical design has revolved, and it is gratifying to know that real advances have been made toward higher stable friction coefficients, coupled with longer life and greater heat resistance. These improved linings were developed for heavy duty buses and trucks, where demands for smaller tires necessitated a reduction in brake diameter. So, borrowing from the field of truck designers' experience, it is now possible to obtain a very satisfactory performance from smaller brakes, provided they are of efficient design, and are used with good wheels. Figure 3 is useful in estimating brake performance from crane characteristics.

In applying brakes it is not sufficient to consider only the leverage ratios involved, for like other machines, it is the ratio of the work input to the work output which determines brake efficiency. In other words, a brake which has little spring (requiring but a small amount of movement to apply pressure to the wheel), and which delivers the greatest friction with the lowest input pressure, will show the highest efficiency. Too high a leverage results in greater spring and excess movement to bring the shoes into drum contact, so that trick mechanisms often fail to demonstrate superior performance. Sloppy fits waste movement, as they require greater shoe clearances, with a corresponding reduction in the available useful input. Higher inputs can be

realized by maintaining the minimum lining clearance permitted by the brake design. Efficiencies will likewise be higher if hard linings are used, and firmly applied to the brake shoes. Careful consideration of the input requirements is necessary to fully utilize the improved linings now on the market—physically represented by increased output from smaller units.

Work input considerations are of relatively great importance only because the brakes on cranes are customarily applied by the human machine and not by motor power. The operator must be able to furnish the input requirements without fatigue, meaning that the designer has distinct limitations. First, foot travel in excess of about nine inches is undesirable, as the whole leg (weighing say 30–50 lb. with a heavy shoe) must be raised the whole nine inches, consuming in the neighborhood of 30 ft.-lb. with every brake application. The second consideration is that the pedal pressure must not be too high—say a maximum of 40 lb. at nine in., and about 90 lb. at three in. (end of the stroke with due allowance for lining wear). Fortunately for the designer, the toggle action of the knee permits higher pedal pressures with the leg extended.

Taking both high and low limits together and averaging the pressures given, shows a human limitation on

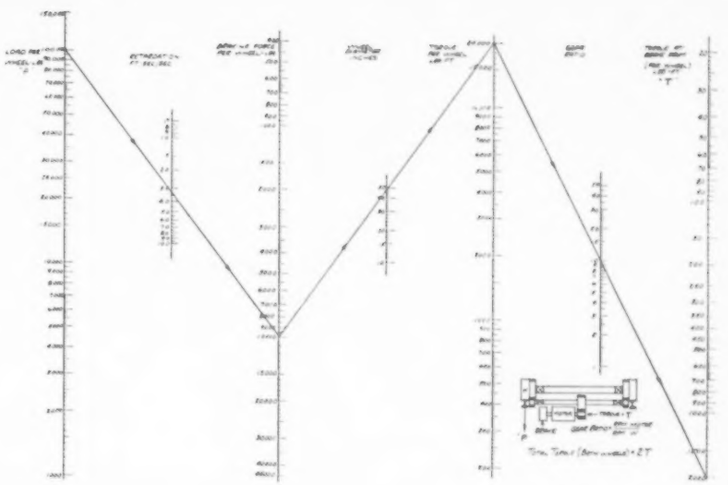
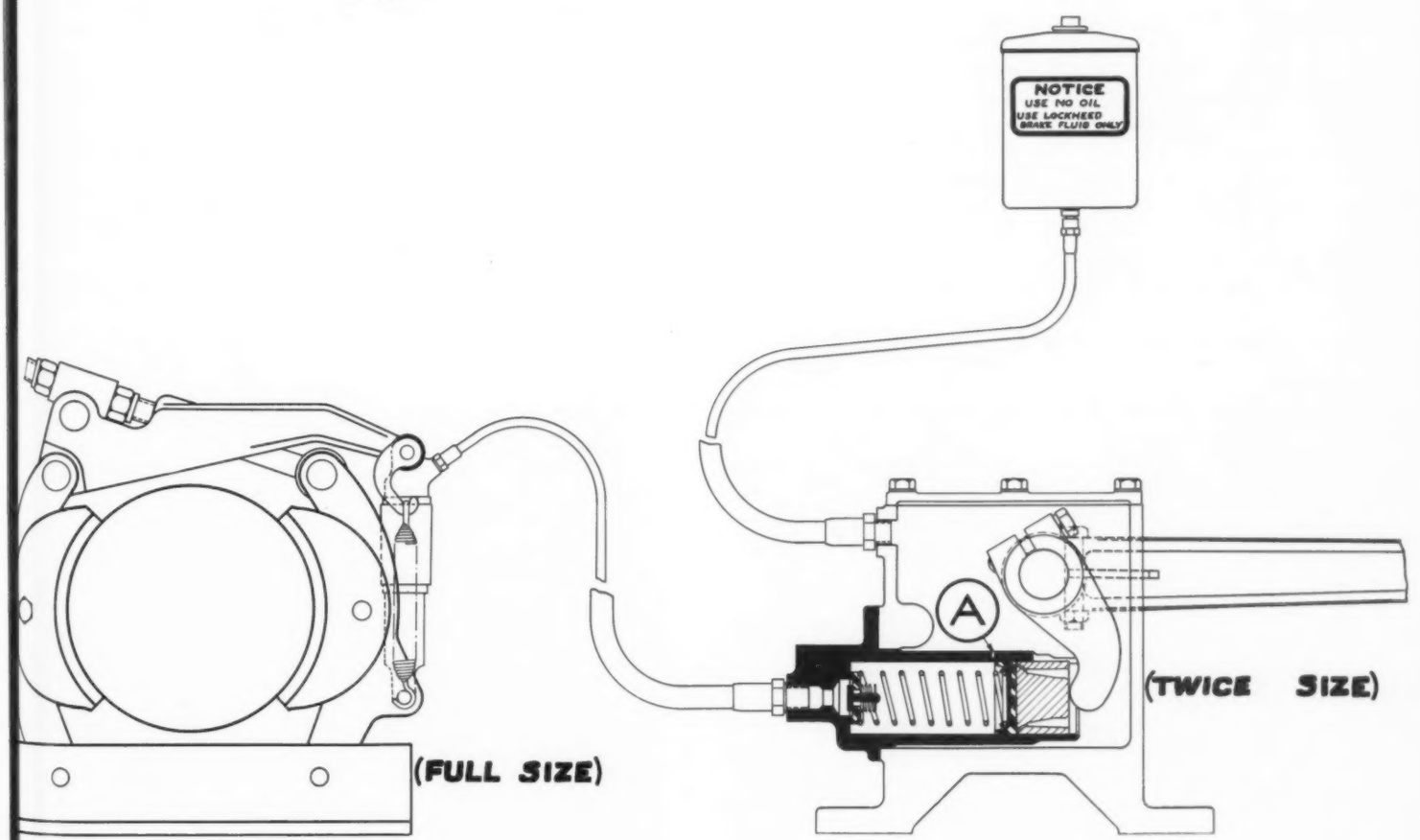


FIGURE 3—Brake torque chart for estimating crane brake performance.

FIGURE 4—Typical hook-up for hydraulic brake on crane bridge.



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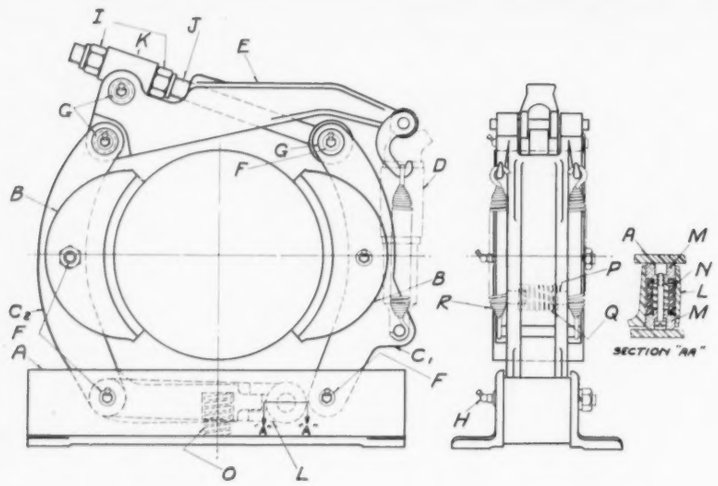

FIGURE 5—Sketch showing assembly for hydraulic brake on crane bridge.

input in the neighborhood of 35 ft.-lb.. although naturally subject to considerable variation.

To use a high pedal ratio in order to reduce pedal effort makes a jumping-jack of the operator, and it is most essential to conserve those 35-odd ft.-lb. available, for it must be remembered that to this figure some 30 ft.-lb. of leg lifting is required if a nine in. pedal height is maintained.

With the work input requirements fixed for a given brake, and the available operator-effort limited, there remains the necessity for efficiently transmitting leg-action into brake-action. Because the operator is usually stationed a considerable distance from the brake, shafting, bell-cranks, levers and bearings will be required, if purely mechanical devices are chosen as the actuating means. Bearing friction can be minimized by providing adequately lubricated, generous bearing surfaces, and by mounting multiplication levers close to the brake unit. From the standpoint of shaft deflection it is also desirable to use no multiplication levers except at the brake proper—and of course all shafts must be heavy enough to transmit torque with minimum deflection.

In latter years, as in automobile design (where inputs are likewise limited) hydraulic actuation has been found most efficient for transmitting work to remote points. Losses in the hydraulic system are confined to relatively insignificant pressure loss due to friction, and almost negligible displacement loss due to expansion. Because these losses usually amount to less than encountered in a single shaft and bearing, the reason for high overall efficiency is apparent. Furthermore, installation is greatly simplified, to obvious advantage.

Because of the relative newness of hydraulic actuation for industrial braking purposes, it will not be amiss to briefly explain how it operated.

In substance, pressure applied to the operating lever displaces the master-cylinder piston—forcing fluid through the line to the brake actuating cylinder (see illustration). As the shoes contact the brake wheel, resistance is increased, and pressure sufficient to produce the required retardation is built up in the system. This phase of the brake's operation is generally understood.

Obviously, the entire system has to be kept full of liquid at all times, if full efficiency is expected. This is accomplished by providing a bleed hole ("A"), which is closed as soon as the master piston has traveled but a small distance, but is otherwise open to permit expansion and contraction of the liquid due to temperature changes.

A two-way outlet valve is located at the end of the master cylinder, which functions to retain a residual pressure of 8–12 lb. in the lines, preventing the column of liquid from draining back into the master cylinder.

A reserve tank is installed at a point near the brake, to further maintain residual pressure, and to provide excess fluid when "bleeding" air from the lines, etc.

It may also be of interest to examine a representative drawing typical of a series of hydraulic bridge brakes designed in accordance with the above principles. In Figure 5, a welded angle-steel base A, supports the brake meahcnism. Two malleable iron shoes, B,B, with bolted-on friction linings, are mounted on hinge arms, C,C. The hinge arms are squeezed together by movement of the wheel cylinder D, actuating the operating lever E, and its associated parts. All pins, F, are pack-hardened and ground, and have serrated ends with lock nuts, which prevent oscillation, thus confirming wear to the ample bearing surfaces provided. Where necessary, replaceable bushings G are fitted. All bearings carry grease fittings H.

Self-locking nuts I on screw J, clamping knuckle K, are part of the simplified adjustment. (It is only necessary to clamp the shoes to the drum with the outer nut, then back-off two turns, lock the inner nut, and adjustment is complete.) Simplified adjustment is possible because shoe clearance is automatically maintained by a slotted arm L frictionally engaging the base A, through plugs M, expanded by spring N. This arm is moved downward by a tongued extension of arm C_2 (which fits in the slot of arm 2) when the shoe clearance is taken up. Release of the applied pressure permits C_2 to move under the influence of spring O, to stop against arm L in a pre-determined position. If the proper adjustment has been made, both shoes clear the drum, and are automatically held clear by repetition of the cycle just described. Plugs P, in each shoe, are spring applied against the arms C_1, C_2 to hold the shoes against rubbing on the drum from their own weight. Return of the operating cylinder is quickly effected by a pair of springs R, arranged to avoid side thrust and consequent binding.

In this design clearances are held very close by an extremely simple adjustment, and practice has demonstrated that very much less pedal movement is required than for earlier types.

While the foregoing material does not pretend to cover all phases of the bridge-brake problem, it is a start in that direction, and may be of value in acquainting those interested with the developments in this field.

RUSTLESS IRON AND STEEL CORPORATION COMPLETES VAST EXPANSION PROGRAM

▲ Rustless Iron and Steel Corporation recently held an open house inspection of a new $2,500,000 addition to the only manufacturing establishment in the world which is devoted exclusively to the production of stainless steel. This Baltimore, Maryland, plant employs unique processes which include every step in the manufacture of stainless steel from the virgin chrome ore to the finished product. The event marked the completion of an expansion program inaugurated by the management in 1935 and intended for long-range accomplishment but which has been shortened to meet the requirements of increased market demand for the corporation's products.

A $1,500,000 plant addition by Rustless in 1937 increased ingot capacity from 20,000 to 40,000 tons a year. This capacity was shortly found to be inadequate, and the current expansion recently completed has increased capacity to 75,000 tons a year. Operations have been stepped up to the point where the number of employees has increased in the past two years from 450 to more than 1400.

The program completed involved the demolition of practically all former buildings to make way for larger and more modern industrial structures, and the plant has been almost entirely equipped with new machinery.

When the management initiated its long-range expansion program in 1935 the corporation's principal producing equipment consisted of three six-ton electric furnaces, a 20 in. rolling mill and a five-stand 9 in. bar and rod mill. In that year the company had a melting capacity of 20,000 tons of ingots.

Melting facilities were changed in 1935 from the original three six-ton furnaces to three of twelve tons each. Two sixteen ton furnaces were added in 1940 which increased the annual melting capacity to 75,000 tons, nearly four times the capacity of five years ago and almost twice the capacity of 1937. To the 20 in. and 9 in. mills which the corporation had in 1935 was added a 12 in. bar mill in 1937. A 28 in. mill has been added in 1940 to produce billets, sheet bars and slabs; as well as a three-stand 12 in. mill and a seven-stand 9 in. mill for the production of bars or coils.

Equipment added to the cold finishing department has more than doubled production, not including a new cold roll mill for the production of special bar and wire shapes to meet present-day conditions and customer requirements. Finished bar and wire ca-

This specially designed charging machine withdraws hot ingots from the heating furnace and swings them to the mill table at right center.

One of the five electric furnaces in the melt shop is being tapped in the view shown below.

pacity is now about 1500 tons a month, between four and five times the capacity in 1935. Other operations have been increased proportionately.

Laboratory facilities have been expanded considerably in the last five years. An uninterrupted program of research also is carried on to explore the ultimate possibilities of stainless alloys and increase their use-value and serviceability to industry.

Until recently stainless steel was considered a part of the general steel industry, but its production has developed so rapidly in the last few years that it now stands apart as a separate industry in its own right. Growth in the ingot production of stainless steel in the period 1934 to 1940 was three-fold, while the production at Rustless was increased nearly ten-fold. This is based on data published by the American Iron and Steel Institute.

SHARON STEEL BUILDS NEW ELECTRIC FURNACE

▲ Sharon Steel Corporation plans a $1,000,000 improvement program for the Youngstown district of that company. Improvements will include a 20-ton electric furnace to be built at the plant at Lowellville, Ohio, and a four-high cold reversing mill to be built at the Sharon, Pennsylvania, rolling mill plant.

FIVE MILLION DOLLAR EXPANSION AT ARMCO

▲ American Rolling Mill Company has announced an important plant improvement program totaling over $5,000,000 for the purpose of effecting substantial operating economies and increasing production. Most important item in the program is a four-stand, four-high, 54 in. cold reduction mill at the Middletown, Ohio, plant. It is estimated to cost $3,800,000 including the necessary buildings and auxiliary equipment. Other major items in the improvement program include constructing 25 additional by-product coke ovens and other improvements at the company's Hamilton, Ohio, plant costing $1,500,000 and a new soaking pit at Middletown.

The cold reduction mill to be installed was designed and built by the Mesta Machine Company and it is estimated that the new mill will start operations in from eight to ten months. Included in the estimated costs are additional furnace capacity for heat treating and tempering steel, a storage building 300 ft. by 380 ft., a pickler building 25 ft. by 330 ft., and a roll shop 80 ft. by 200 ft. The storage building will also house the shears, rewinding and other equipment for cutting and handling coils as well as warehousing coils which are being processed.

Armco already operates a cold reduction mill in Middletown for rolling sheets 20 gauge and heavier, and in widths up to 75 in. The new mill will supplement it by rolling sheets up to 48 in. wide and as light as 30 gauge. Each of its four stands of rolls will be

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driven by large motors, two of which are 2,000 hp. and two of 2,500 hp. It will roll steel coils at a maximum delivery speed of 2,100 ft. per min.

Construction of the 54 in. mill will not permit the company to make both heavy and light gauge flat-rolled products without reheating. Due to fuel savings and other efficiencies, production costs should be reduced substantially.

It has also been decided to construct a new soaking pit at the Middletown plant for heating ingots prior to rolling. The new soaking pit, which was designed and built by the Surface Combustion Corporation, will increase the rolling capacity of the company's hot strip mill during periods of peak demand.

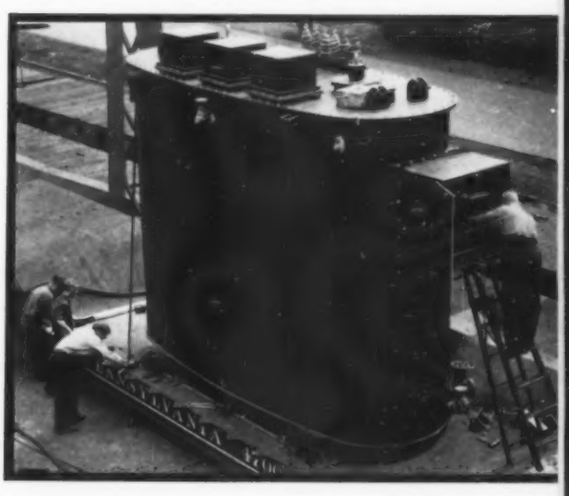
Plans have also been made for enlarging the Middletown plant's open hearth production, although no new furnaces will be built. The mould yard is being enlarged and new trackage will be built so additional hot pig iron, scrap, and other raw materials may be transported to the open hearth furnaces. The capacity of the fuel storage is also being enlarged, and a hot lime soda process water softener of 50,000 gal. per hr. capacity is being installed to increase the supply of treated water.

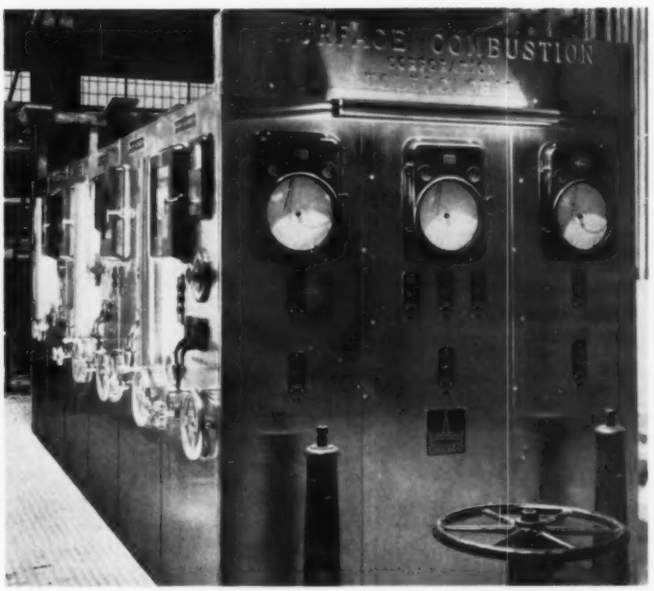
About $1,500,000 is being spent to improve the Armco blast furnace plant at Hamilton, Ohio. Twenty-five new by-product coke ovens, designed by the Koppers Company, will be added to the present battery of 60 to increase the coke supply. The capacity of the ore yard will be increased by 100,000 tons, and a modern turboblower is being installed. Recently one of the Hamilton blast furnaces was enlarged to increase its capacity by about 350 tons per day.

PRODUCTION RECORD SET BY TRANSFORMER FIRM

▲ What is believed to be a record in speed was accomplished by Pennsylvania Transformer Company, recently, when one of the largest furnace transformers in the country was shipped from their plant September 20, 1940. The order for this huge transformer was received August 16, so that only thirty-five calendar days elapsed from the time of actual receipt of order. Every part of this unit was designed and built in the Pennsylvania factory. The transformer has

Thirty-five days after receipt of the order, this transformer, one of the largest in the country, was loaded for shipment from the factory, setting a new production record.





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Fact is, you are one of them yourself . . . and so we wish you could stand in front of this handsome, stainless steel panel board that houses the Askania Control of two 4-zone oil—or gas—fired slab heating furnaces.

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"IF IT CAN BE MEASURED—WE CAN CONTROL IT"

a physical capacity of 20,000 kva., 3 phase, 25 cycles, and will supply power to a furnace installed in the plant of a leading industrial concern.

WIRE FIRM MODERNIZES WORCESTER STEEL WORKS

▲ American Steel and Wire Company recently announced the largest rehabilitation and improvement program ever undertaken by that company in Worcester, Massachusetts. The program includes four additions to the company's present South Works and complete re-arrangement and improvements which will result in one of the most efficient wire mills in the United States.

The largest of the new buildings 1035 ft. x 500 ft. will house the manufacture of round and shaped wire and strip. This will permit manufacture of all the company's steel products now produced in Worcester under one roof. Connected with this mill will be a two story office building and physical laboratory.

The present spring mill at South Works will be re-arranged and the rail bond and flat coiled spring operations will be consolidated in a two story building, 186 ft. x 120 ft. The company's copper facilities at Worcester also will be re-arranged and consolidated in two new buildings. One will include the copper stranding operations and warehouse, and the other, one story—400 ft. x 80 ft.—will include copper drawing, cleaning, annealing, cold rolling and tinning operations.

When the new mill is completed the present North Works and Central Works will be abandoned and the present wire drawing building at South Works demolished.

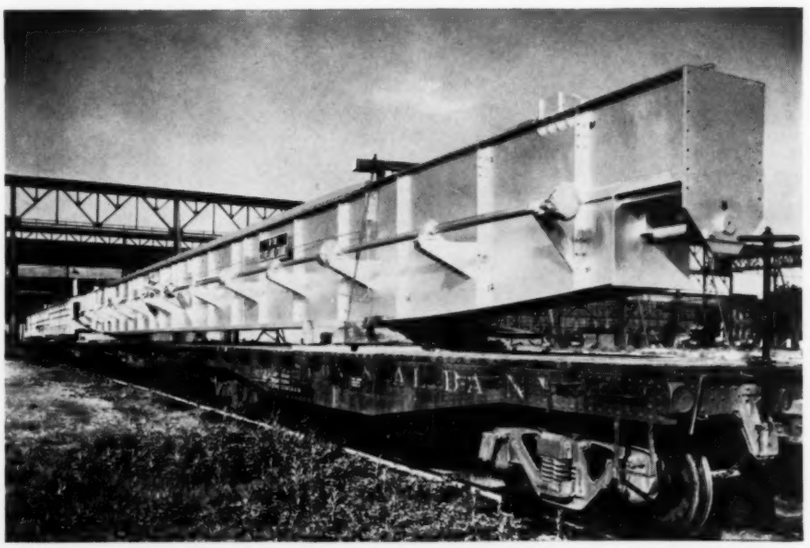
DEVELOP REVOLUTIONARY SHELL FORGING PROCESS

▲ The development of a new and revolutionary hydraulic shell forging press, designed upon the principle of making a complete shell forging in one operation without extrusion of the steel during the piercing of the billet, is announced by United Engineering and Foundry Company.

By making the complete shell forging in one operation, a second step, or draw press operation, is completely eliminated. Greater accuracy and concentricity is obtained and minimum amount of metal is required in the billet.

A heated billet is inserted vertically in a book-type split and hinged die. A control lever sets in motion two hydraulic horizontal rams that operate the wedges for closing the water-cooled die around the billet and at the same time a lower vertical supporting ram moves upward. Actuated by the top and main hydraulic ram the punch unit moves downward and into the billet for the piercing operation.

As the punch progresses into the billet, the metal flows sideward and downward beside and ahead of the punch tip. When the punch has reached the bottom of its stroke and



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EVERY CRANE ALL-WELDED

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Cleveland Crane has the confidence of steel mill engineers. From experience with all kinds of cranes, they know that no crane can excel an all-welded crane (meaning of course, a crane welded throughout, including the girders) for *hard* day in and day out service.

That's why big orders for Cleveland all-welded Cranes are placed without hesitation.

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CLEVELAND CRANES
ALL-WELDED OVERHEAD TRAVELING CRANES
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the cavity of the shell forging formed, a lever is automatically tripped for the return stroke. This also opens the die when the punch has retracted.

Only about two seconds are required for the actual piercing operation and the press is designed to forge any shell from 75 mm. to 155 mm., inclusive.

NEW HIGH LIFT TRUCK FOR CONGESTED AREAS

▲ The latest addition to the line of industrial trucks manufactured by the Baker Industrial Truck Division of the Baker-Raulang Company, Cleveland, is its new high lift truck which is available in 4000-lb. capacity. This new model has been designed for operation in narrow aisles and congested areas, and has the following features: efficient hydraulic lift; lighter weight without sacrifice of strength or ruggedness; shorter overall length; accessibility—all controls and hoists come grouped in convenient control panel; and absolute control of hoisting and lowering speeds.

This new high lift truck has a standard telescoping lift of 119 in. and is hydraulically operated by a gear pump driven by its own motor.

Lifting is accomplished by a hydraulic system incorporating the latest developments in this type of equipment consisting of a gear pump driven by its own motor and connected to the lifting cylinder through suitable control and release valves. The control valves are of the metering type, permitting absolute control of the speed at all times.

The standard simple lift is 60 in. The standard telescoping lift is 119 in. This new model will operate in aisles 61 in. wide.

COMPOUND PUMP UNIT USES SPECIAL MOTOR

▲ A new "electrifugal" pump, which is an all-in-one centrifugal pump especially designed as a complete pump and motor unit on one shaft and one housing, has been developed by the Allis-Chalmers Manufacturing Company. Instead of using standard motors having special end housings and shaft extensions, the "electrifugal" pumping unit has a special motor with a one piece cast iron motor yoke and pump bracket. The feet are cast integral with the housing and bracket and extend under the entire unit instead of under the motor only.

The special motor design, with copper bearing steel cover, meets NEMA specifications for splash proof motors. Totally enclosed, fan cooled

(*Please turn to page 85*)

Spindle Design *and* Construction Contribute *to* Close Accuracy *and* Fine Finish *of*

FARREL ROLL GRINDERS

Contributing substantially to the high precision and fine finish for which Farrel Heavy Duty Roll Grinders are noted are the construction and mounting of the grinding wheel spindle. Heavy, rigid and smooth-running, the wheel spindle is one of the outstanding design features of the machine.

For rigidity under working load the spindle is made as large as possible in diameter. It is made of high grade alloy steel, carefully machined, heat treated and ground to extremely close limits of accuracy. It is mounted in special bronze alloy, or steel-backed, babbit-lined bearings of the sleeve type, split longitudinally to fit a wedge by means of which the running clearance is closely controlled.

Thrust on the spindle is taken by a hardened steel disc and two babbitted rings at the center of the spindle, which permits free expansion of the spindle bearings and minimizes deflection of the spindle under thrust load.

Vibrationless, multiple V-belt drive, continuous automatic lubrication and other Farrel features combine to assure the smooth-running spindle needed to deliver continuously close accuracy and fine finish of ground rolls.

For full particulars of the Farrel Heavy Duty Roll Grinder send for a copy of Bulletin No. 111

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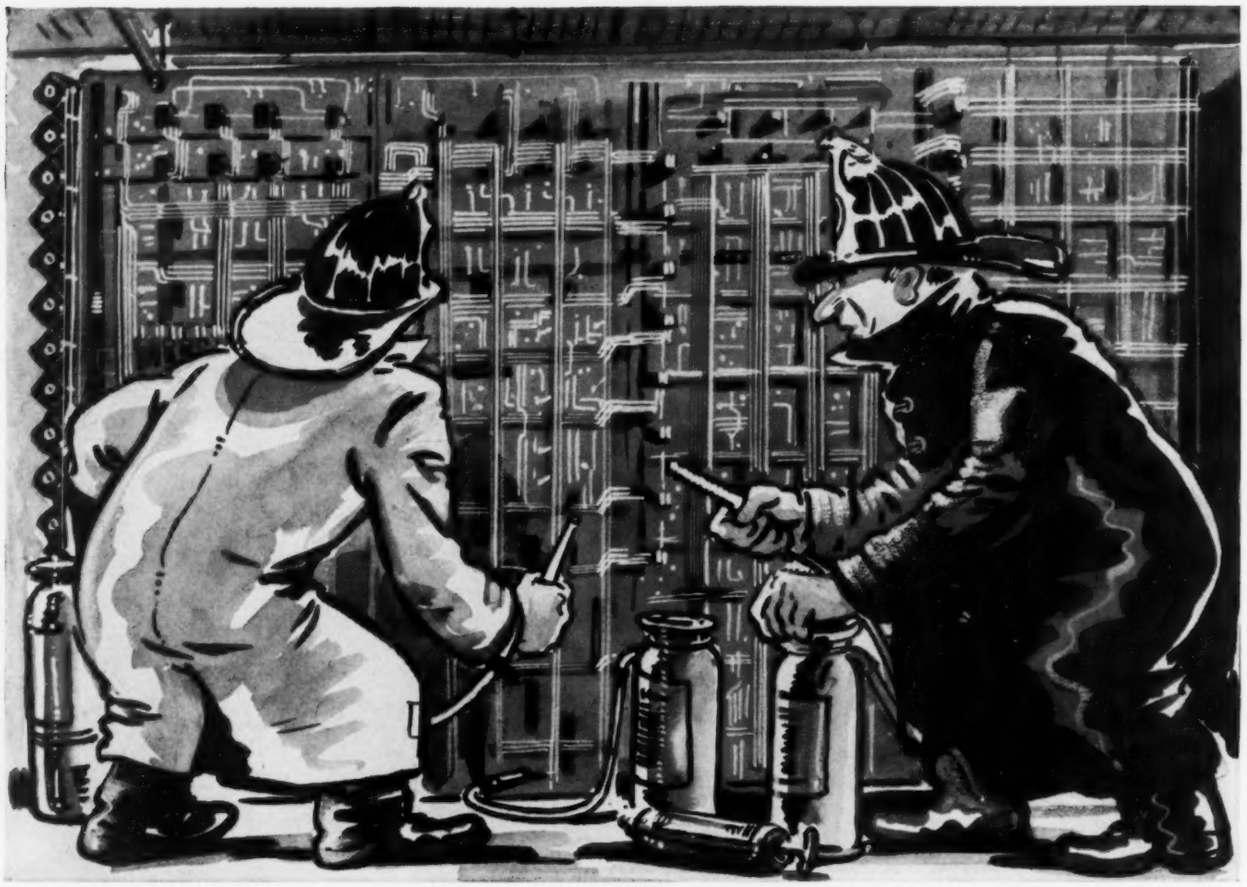

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You could take a lot of the danger out of electrical fire hazards by posting alert fire-laddies around your switchboards, grid connections, control, lighting and power circuits . . . but the easiest and most economical way to prevent wire-fires is to use Rockbestos fireproof and heatproof permanently insulated wires.

When you stop to consider that every circuit of ordinary wire in your plant offers an easy pathway for the rapid spread of fire, and particularly heavily grouped circuits, then you can appreciate the need for a wire that is *positively fireproof*. One that won't burn, smolder, or even carry flame, under arc conditions or any other condition. And you'll get it if you order Rockbestos because our entire line of wires, cables and cords is insulated with *fireproof asbestos*.

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You'll get other plus values too, because Rockbestos wires are resistant to heat, moisture, oil, grease, corrosive fumes and alkalies, and they won't dry out, crack, rot, bloom or swell like other insulated wires.

For wiring jobs where you want positive protection against fire, or resistance to high temperatures and other insulation-destroying agents, select your requirements from the Rockbestos line of permanently insulated wires, cables and cords. We make more than 100 standard constructions, every one of which has made good in service under the severe operating conditions it was designed to meet. Our No. 10-E Catalog tells all about them. Send for it. Rockbestos Products Corporation, 975 Nicoll St., New Haven, Conn.

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ROCKBESTOS — *the wire with permanent insulation*

A typical construction — Table RF, 600 volt Rockbestos All-Asbestos Flexible Apparatus Cable — one of 118 different permanently insulated wires and cables.

(*Continued from page 84*)

motors and explosion proof motors are also available in this new pump.

In splash proof construction the pump is now available in sizes from 1 hp. to 10 hp., inclusive, at 3500 rpm. and from ¾ hp. to 7½ hp., inclusive, at 1750 rpm. for heads up to 160 ft. Larger sizes of this type are in the process of design.

BUILD ELECTRIC FURNACE AT CANTON STEEL PLANT

▲ Timken Roller Bearing Company has announced placement of an order for a new 60-ton electric furnace to meet increased demands for aircraft steel in connection with the national defense program. The total expenditure involved is $500,000.

This, the second 60-ton electric furnace to be installed by Timken in recent months, will boost their electric steel production from 21,000 to 27,000 net tons per month, when it is placed in operation about February 15.

The world's largest electric furnace, a 100-ton producer, is included in this company's battery of six electric furnaces. One of the furnaces has a 10-ton capacity, two have a 35-ton capacity and two have 60-ton capacity. Three open hearth furnaces add 15,000 additional tons per month to the Timken steel mill's output.

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GOOD furnace control requires an accurate record of draft, pressure or differential, as given by Hays Series OT Supersensitive Draft Recorders.

These instruments give a correct indication of furnace atmospheres and a permanent record of pressures and drafts at vital points as guides for most effective operation.

Hays OT Draft Recorders give two draft values, two pressure values, two differential values or a combination of any two of these three values. Built stout enough for Steel Mill service yet sensitive enough to register accurately in increments of .0025 inches water.

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ANNOUNCE LINE STARTER FOR INDUCTION MOTORS

▲ Magnetic motor starters designed for across-the-line starting of single and polyphase squirrel cage induction motors, and as primary control for wound rotor induction motors, have been announced by the Colt's Patent Fire Arms Manufacturing Company.

One important feature of the new magnetic starters is the three-point ball-bearing suspension of the movable electro-magnet, assuring a floating action that minimizes friction and guides contacts smoothly into correct alignment. The magnet operates vertically and cannot close accidentally by vibration or shock.

GALVANOMETER SUITABLE FOR MANY APPLICATIONS

▲ A new galvanometer, more than three times as sensitive as previous galvanometers, has been announced by the General Electric Company. The new instrument is suitable for many applications where wall types were heretofore frequently necessary. Applications include temporary setups for factory and laboratory testing; permanent installations for testing instruments, material, and apparatus for production, where rapid readings and minimum fatigue to the operator are essential; and for bridges and other apparatus requiring an external galvanometer.

The extreme sensitivity of this instrument is largely obtained by an

arrangement of two fixed mirrors mounted inside the case, one on each end, thus increasing the effective light beam length. This gain in sensitivity is obtained at no sacrifice in strudiness.

The galvanometer element and optical system are mounted in an attractive case of cast-aluminum alloy. The scale, of translucent compound, is double-marked 50-0-50 and 0-100 in 1-mm. divisions. The light source is provided by an automobile-type lamp, easily replaceable, which provides a high-intensity spot with hairline index, so that the scale can be read quickly, accurately, and with minimum eye fatigue. An external adjuster is provided for setting the element on zero. Alnico magnets are employed and the coils are of "Formex" wire, one of the latest products of General Electric research.

NEW WICK FEED OILERS HAVE GOOD VISIBILITY

▲ A new visible and unbreakable wick feed oiler has been developed by the Trico Fuse Manufacturing Company. The oiler is intended for economical use on machinery which operates intermittently. At the top of the oiler is a lever which when in a vertical position starts the feed, and when "flipped" to the side shuts off the feed. There is no flooding and waste of oil when the machine is idle. The oil supply is always visible and the crystal clear unbreakable reservoir tells at a glance when refilling is necessary.

When oiler is filled above the point where the wick enters the center tube, the surplus oil drains into the bearings, thus finishing it thoroughly. When the oil recedes to the opening in the tube, the wick automatically feeds oil to the bearing by capillary action. Dirt or sediment cannot enter the bearing. This oiler is fifty per cent lighter in weight and there are no gaskets to leak as the unbreakable bottle is cemented and roll-clinched to the heavy brass base. All metal parts are bright chromium plated for beauty and easy cleaning.

The shut-off lever is hidden by the streamlined dome. The dust-proof, self-closing filler cap has an extended lip on the cover so that it can be easily raised with the spout of the oil can. The wick is wire reinforced for rigidity.

RUST PREVENTIVES FOR NON-DRYING PRODUCTS

▲ A new series of rust preventives, including liquid rust-inhibiting non-drying products built from a newly developed base. This series also includes grease-type (non-drying) products; drying, hard-film products, semi-drying (waxy film) and soluble rust preventives.

The new basic material provides a homogeneous film non-crystalline, with a high degree of cohesion as well as adhesion to metals. It wets out the surface rapidly making a "non-tacky" unbroken film impervious to atmospheric conditions. It is described as the greatest advance in protection against ravages of rust since E. F. Houghton and Company first began making rust preventives in 1869.

Products are provided to meet government and industrial specifications. A new method of accelerated testing of rust prevention has also been developed to aid industry in selecting the most adequate anti-corrosive covering.

PHOTO-ELECTRODE UNIT AVOIDS FLAME FAILURE

▲ A safe and relable method of protection against failure of oil, powdered coal, and luminous gas flames by the recently developed "Protectoglo" photo-electrode system is announced by the Brown Instrument Company. The photo-electric principle used in the new unit avoids the necessity of placing a metallic electrode in the high temperature flames, but retains the quick response feature which is characteristic of thermionic systems. The function and operation of the photo-electrode is quite similar to the familiar Brown flame-electrode "Protectoglo" system. The principle difference lies in its method of flame detection. Instead of using a metallic electrode the new photo-electrode system detects the presence of a flame by means of the light emitted when a burner is in normal operation.

The heart of the system is essentially a light-sensitive device. In operation, the light from the flame is directed upon a photo-cell. This results in a small flow of current through the cell which is amplified by a vacuum tube circuit until of sufficient magnitude to close a relay. Through the relay contacts, the main fuel valve is energized.

UNITED STATES STEEL WILL EXPAND IN SOUTH

▲ Tennessee Coal, Iron and Railroad Company, a subsidiary of the United States Steel Corporation, has announced a broad program involving an expansion of the steel making and finishing facilities at Birmingham, Alabama. The new facilities, when installed, will result in approximately a twenty per cent increase in the ingot capacity, a little more than 400,000 tons, and a corresponding increase in the finishing capacity of this company.

These improvements, which will commence at once, but which by reason of their nature, require about eighteen months to complete, will then place the Tennessee Coal, Iron and Railroad Company in a materially improved position to carry out any obligation which may be required as a consequence of the national defense program.

The improvements comprise the following important features: construction of an additional battery of coke ovens and an additional blast furnace at Fairfield steel works; improvements to the existing open hearth furnaces; development of ore and coal mining facilities to meet requirements of the new blast furnace, additions and improvements to the plate mill at Fairfield, including a new 140 in., four-high plate mill; additional wire drawing, galvanizing and finishing facilities at the Fairfield wire works; and additional processing and finishing facilities at the Fairfield sheet mill.

These changes will necessitate enlarged operations at the ore mines, coal mines, and quarries, including mechanical equipment, additional transportation facilities, and to the extent, necessary enlargements and improvements in the steam and electric power plants and the water supply system.

An increase of 400,000 net tons will bring ingot capacity of this company to around 2,650,000 net tons.

STANDARDIZE EQUIPMENT FOR CIRCUIT BREAKERS

▲ A new standardized automatic reclosing equipment for use with small outdoor oil circuit breakers of from 50,000 to 250,000 kva. interrupting

capacity, 400–600 amperes at 7500 volts, and 600–1200 amperes at 15,000 volts a-c., is announced by the Westinghouse Electric and Manufacturing Company.

These new reclosing equipments were specially designed to meet a growing need. Certain breakers applications must always be supplied with reclosers, but slight variations, due to individual requirements, made their manufacture extremely costly. This new equipment will meet 95 per cent of all application requirements, and at a much lower price than made to order apparatus.

The new equipments utilize the time tried RC recloser, and include complete breaker accessories, such as current transformers, breaker trip coils, rectox, etc., and a control cabinet and panel completely wired and mounted on the breaker frame. They are used in conjunction with the Westinghouse types FO-22-A and GO oil breakers.

NEW LITERATURE

▲ Blaw-Knox Company has prepared a catalogue on its series of two-line lever arm clamshell buckets. A total of 242 individual bucket specifications are involved. These 242 buckets are listed in a consolidated table according to rated capacity, ranging from ⅛ to 7½ cu. yd. This table records the service classification of each bucket, lists dimensions and physical data, and gives, in many instances, the approximate cu. ft. performance on different classes of material.

Catalogue No. 1757 may be obtained from the Blaw-Knox Company, Pittsburgh, Pennsylvania, as well as the recently issued "Buckets for Single Drum Hoists" No. 1696, which is also available.

▲ Hagan Corporation is printing for distribution a timely new booklet entitled: "Put Your Power Plant on a Preparedness Basis." This booklet portrays many of the conditions prevalent during the last World War, and gives many reasons for the need for industry to look into its power plant problems.

In conjunction with this booklet, a supplementary pamphlet: "Seventeen Case Histories from the Files of Hall Laboratories" is being issued by Hall Laboratories, Inc. This material was extracted directly from the client files by Dr. R. E. Hall, director, and is presented in a brief, straightforward and personal manner. It provides a clear insight into various water problems existing in representative plants throughout the country and relates how these problems were solved through Hall System of boiler water conditioning.

Copies of both publications may be obtained by writing to the Hagan Corporation, Bowman Building, Pittsburgh, Pennsylvania.

▲ Jefferson Electric Company has just printed a new 12-page bulletin on power circuit air-cooled transformers. Illustrations of the various types are shown as well as suggestions for applications, winding diagrams, and specification and dimension tables.

Bulletin 401-PCT may be obtained by writing to the Jefferson Electric Company, Bellwood, Illinois.

▲ Koppers Company has arranged a check-list of the available literature describing the various activities of that company. Nearly fifty separate publications are listed in a convenient manner so that persons desiring a particular type of publication may so indicate.

Copies of this list will be sent upon written application to the publicity department of the Koppers Company, Koppers Building, Pittsburgh, Pennsylvania.

▲ Askania Regulator Company has published a book entitled "Evidence —that Askania is your kind of control." It shows how the Askania jet pipe regulator is being used to control furnaces in the iron and steel industry. Engineers and plant operators interested in seeing how other plants have solved the problem of automatic control can secure a copy of this booklet without obligation by writing to the Askania Regulator Company, 1603 South Michigan Avenue, Chicago, Illinois. Readers who are not familiar with the Askania hydraulic jet pipe principle should also ask for Bulletin No. 100.

▲ Trumbull Electric Manufacturing Company has available copies of the September "Trumbullist." This issue contains the addition of several new lines, notably: the new "MH" multi-breaker power panel and the announcement of their new motor control centers, both of which are illustrated on the inside front and back covers. Also it contains the addition of three-phase, four wire, 100 ampere "M-2" multi-breakers.

Copies of this catalogue will be sent upon request to the Trumbull Electric Manufacturing Company, Plainville, Connecticut.

▲ Cochrane Corporation has available the following catalogues which may be obtained by writing to Cochrane Corporation, 17th Street and Allegheny Avenue, Philadelphia, Pennsylvania.

Hot Process Softeners: Twelve-page bulletin No. 2059 describes the hot-lime-and-soda feed water treatment when combined with integral deaeration equipment to eliminate dissolved oxygen and other corrosive gases.

Zeolite Softeners: Twenty-four page bulletin No. 2860 consists of a well-written explanation of zeolite softening processes together with complete descriptions of apparatus required. Complete tabulated data and charts permit calculation of list prices for any desired installation.

Deaerating Heaters and Deaerators: "Preventing Corrosion by Zero Oxygen," sixteen-page bulletin (No. 2540) discusses in detail the advantages of minimizing or completely eliminating dissolved oxygen and other gases from boiler feed so as to reduce pitting and corrosion to boiler and auxiliary surfaces and piping.

▲ Leeds and Northrup Company has available a complete publication on thermocouples which not only lists a comprehensive line of assemblies with their parts and accessories, but includes information of general usefulness on the correct choice of couples. Tabulated in easy to use form, this information should be helpful to users as a guide to the best choice of couple for the application at hand.

A copy may be obtained by writing to the Leeds and Northrup Company 4934 Stenton Avenue, Philadelphia, Pennsylvania, and requesting Catalogue N-33-A(6).

▲ Pyle-National Company has published a bulletin on their "QuelArc" heavy duty plugs and receptacles. This bulletin describes the use of these plugs and receptacles with heavy duty portable electric equipment.

Copies of the bulletin may be obtained by writing to the Pyle-National Company, 1334–58 North Kostner Avenue, Chicago, Illinois, requesting Bulletin No. 1140-2.

ITEMS OF INTEREST

Paul B. Tonnar, formerly assistant chief engineer of Corrigan-McKinney Company and chief draftsman of Republic Steel Corporation, has been appointed chief engineer of the Andrews Steel Company, Newport, Kentucky. Mr. Tonnar joined the Andrews company in January, 1940, as mill designer. Previous to his affiliation with the Andrews Company, Mr. Tonnar was connected with the Arthur G. McKee Company.

M. E. Morgan, open hearth, blast furnace and coke works superintendent at Republic Steel Corporation, Gadsden, Alabama, has been promoted to superintendent of blast furnaces and coke works for the Youngstown district of that company.

George Cain, formerly superintendent of open hearths, Republic Steel Corporation, Cleveland, Ohio, is now superintendent of open hearths at the Gulfsteel plant at Gadsden, Alabama.

W. H. Winton has been appointed blast furnace and coke works superintendent at Republic Steel Corporation, Gulfsteel plant, Gadsden, Alabama. Mr. Winton was formerly assistant superintendent of blast furnace and coke works at the Gulfsteel plant in Gadsden.

C. J. Smith was appointed assistant to general superintendent in charge of engineering and maintenance at the Carnegie-Illinois Steel Corporation, Gary, Indiana. Mr. Smith formerly held the position of superintendent of the maintenance division, and previous to that was head of the auxiliary departments.

C. J. SMITH

T. R. Miller, who has been coordinator of standard costs and budget supervisor at the Gary Works of the Carnegie-Illinois Steel Corporation, succeeds Mr. Smith as superintendent of the maintenance division at Gary.

W. S. MacNabb, formerly assistant division superintendent of open hearths and central mills, succeeds T. R. Miller with the title of assistant to general superintendent, coordinator of standard costs and budget supervision at the Gary Works of the Carnegie-Illinois Steel Corporation.

Donald Petersen was appointed superintendent of transportation for the Carnegie-Illinois Steel Corporation's Gary Works.

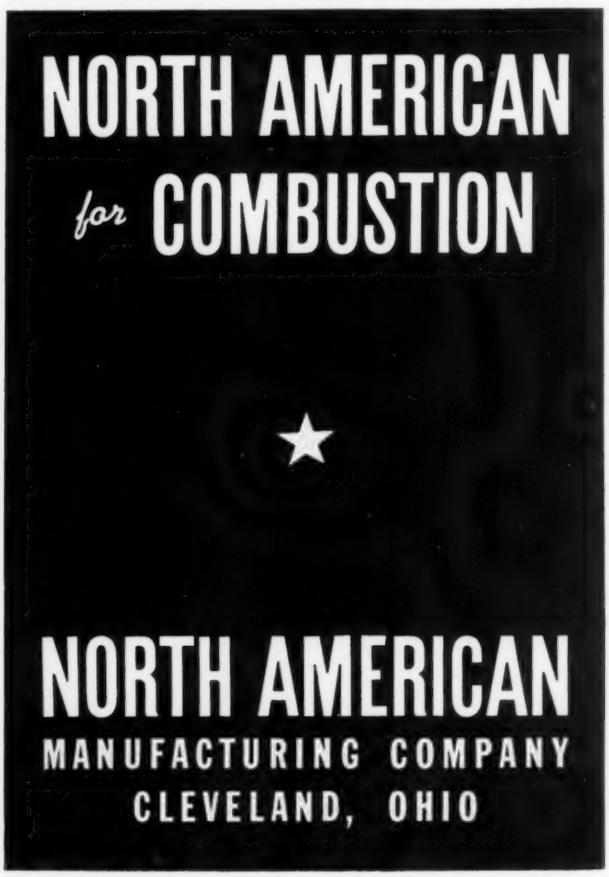
William F. Conlin, division metallurgist in the Cleveland office of the American Steel and Wire Company, has been appointed open hearth superintendent at the steel works in Donora, Pennsylvania.

John A. Slenker, open hearth superintendent at American Steel and Wire Company, Donora, Pennsylvania, was appointed division metallurgist at Cleveland, Ohio.

Henry R. Patterson, assistant general superintendent of the Donora steel and wire works of the American Steel and Wire Company, was moved to the main office in Cleveland as engineer of wire drawing development.

A. F. Ilacqua has been appointed division metallurgist on high carbon steels at the main office of the American Steel and Wire Company, Cleveland, Ohio.

S. J. Horrell has been appointed vice-president of the Power Piping Division of Blaw-Knox Company, Pittsburgh, Pennsylvania. Mr. Horrell became associated with the company in May, 1939, and has been serving this division in the capacity of sales manager. These duties have been increased to include charge of engineering and sales. He has had more than 18 years' experience in piping work.



Dr. Claude L. Clark, formerly of the department of engineering research of the University of Michigan, has joined the metallurgical staff of the Steel and Tube Division of the Timken Roller Bearing Company at Canton, Ohio, as metallurgical development engineer. During the past ten years at the University of Michigan, Dr. Clark has carried on studies of the properties of metals at elevated temperatures, for which Timken has been one of the principal sponsors. He will continue to devote the major portion of his time to the application of high temperature steels for the Steel and Tube Division.

Obituary

T. J. Flaherty, electrical superintendent, American Rolling Mill Company, Ashland, Kentucky, died on November 10, 1940. Mr. Flaherty started to work in 1900 as a helper in the electrical department of the La Belle plant of what is now the Wheeling Steel Corporation, Steubenville, Ohio. He worked as motor inspector for the Colorado Fuel and Iron Company, Pueblo, Colorado, in 1904 and 1905. In 1905 he returned to the La Belle plant where he worked as electrical foreman until 1910.

T. J. FLAHERTY

In 1910 Mr. Flaherty went with the American Rolling Mill Company, Middletown, Ohio, as electrical foreman. He left Armco in 1913, going to the Cambria plant, Bethlehem Steel Company, Johnstown, Pennsylvania, where he became general foreman in the electrical department. Mr. Flaherty joined the American Rolling Mill Company again in 1923, going to the Ashland, Kentucky, plant as assistant electrical superintendent. He was made electrical superintendent in 1925, holding that position until his death.

An active member of the Association of Iron and Steel Engineers since 1915, Mr. Flaherty has freely participated in the various activities of the society, serving as a member of the board of directors in 1939.

H. H. Timken, Sr., chairman of the board of the Timken Roller Bearing Company, died recently in his home at Canton, Ohio. Mr. Timken attended public schools in St. Louis, Missouri, and spent many of his vacation periods in his father's carriage works, where he observed first-hand the processes of manufacturing. Upon his graduation from high school he enrolled in Washington University in St. Louis. He later entered the University of California and was graduated from there with a law degree.

Mr. Timken, with his brother William R. Timken, founded the Timken Roller Bearing Company in Canton, Ohio, in 1898, and production was started with a force of less than fifteen men. After successfully establishing his company's place in the automotive field, and the company expanded, Mr. Timken became interested in the development of new steel alloys. He began making steel in an electric furnace and after months of experimenting and many discouraging setbacks, finally produced an alloy that met his requirements.

H. E. McCoy, head of the H. E. McCoy Company in Pittsburgh, Pennsylvania, died recently. Mr. McCoy had been connected with the iron and steel industry in one way or the other for over fifty years. He began his career as a pattern maker with the Porter Foundry in Pittsburgh, and served in that capacity for other foundries in the Pittsburgh district. Leaving his trade to enter business for himself, Mr. McCoy subsequently formed the McCoy-Brandt Machinery Company, dealing in new and second-hand machine tools. This company was dissolved about 1917, and one of the accounts handled at the time, Baker-Raulang Company, makers of industrial trucks, was retained by Mr. McCoy. He was the first accredited representative of the Baker-Raulang Company, and continued to handle this account in the Pittsburgh district until his death.

H. E. McCOY

F. K. Tribby, chief electrician, Pittsburgh Coke and Iron Company, Pittsburgh, Pennsylvania, died recently. Mr. Tribby was appointed chief electrician of the Neville Island coke and iron plant, at the time it was built and has held that position for fifteen years. Previously, Mr. Tribby had been employed by the Valley Mould and Iron Corporation at the Sharon and Middlesex plants.

To the Younger Men in the Iron and Steel Industry:

The Constitution of the Association of Iron and Steel Engineers was recently revised. Included in the revisions is a classification of membership designed to fill the demands of the younger men in the iron and steel industry. This classification is known as JUNIOR MEMBER.

Under the revised Constitution JUNIOR MEMBERS shall be citizens of the United States or Canada and shall be persons identified with the Iron and Steel Industry, not less than 18 years of age and not eligible to membership under any other classification. After the age of 27 years, one's classification automatically becomes Active or Associate, subject to the approval of the Board of Directors. The yearly dues are changed accordingly and no entrance fee is required to make the transfer.

Annual dues for JUNIOR MEMBERS are five dollars. There is no entrance fee. JUNIOR MEMBERS are entitled to the monthly issue of the *Iron and Steel Engineer,* the official publication of the A. I. S. E., the privilege of attending all district section meetings (in Chicago, Birmingham, Detroit, Cleveland, Philadelphia, and Pittsburgh), the privilege of attending the Annual Convention and Iron and Steel Exposition, the Spring Engineering Conference, and the regular inspection trips to steel plants.

For full particulars please address inquiries to your nearest District Section office:

Birmingham District Section: A. L. Lemon, 106 N. 22nd St., Birmingham, Alabama.

Chicago District Section: A. J. Whitcomb, 310 South Michigan Avenue, Chicago, Illinois.

Cleveland District Section: W. W. Spanagel, 1080 Ivanhoe Road, Cleveland, Ohio.

Detroit District Section: Elmer Weiss, 2755 East Grand Boulevard, Detroit, Michigan.

Philadelphia District Section: Linn O. Morrow, Schaff Building, Philadelphia, Pennsylvania.

Pittsburgh District Section: Brent Wiley, 1010 Empire Building, Pittsburgh, Pennsylvania.

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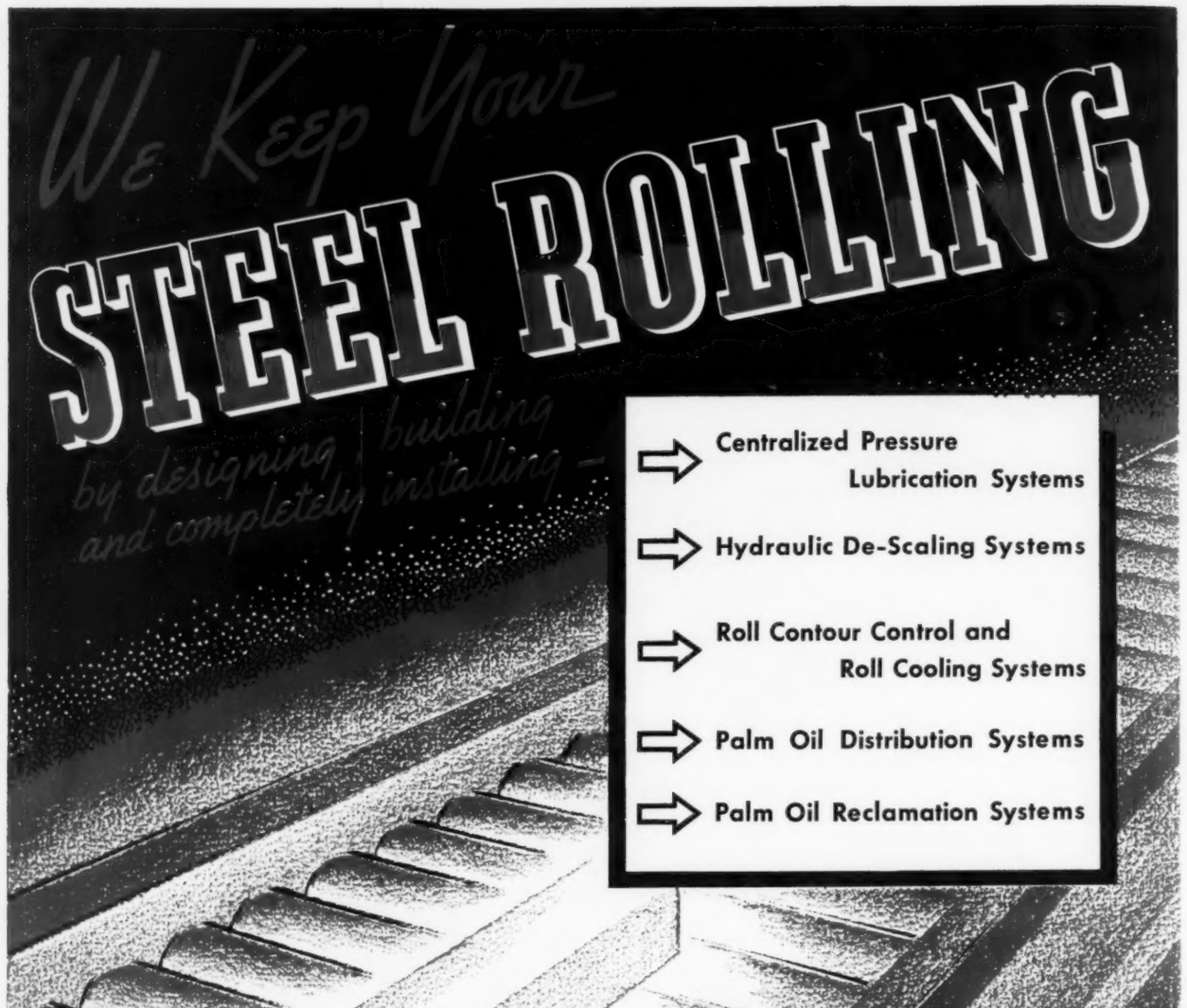

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